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January 3, 1911.

POWER

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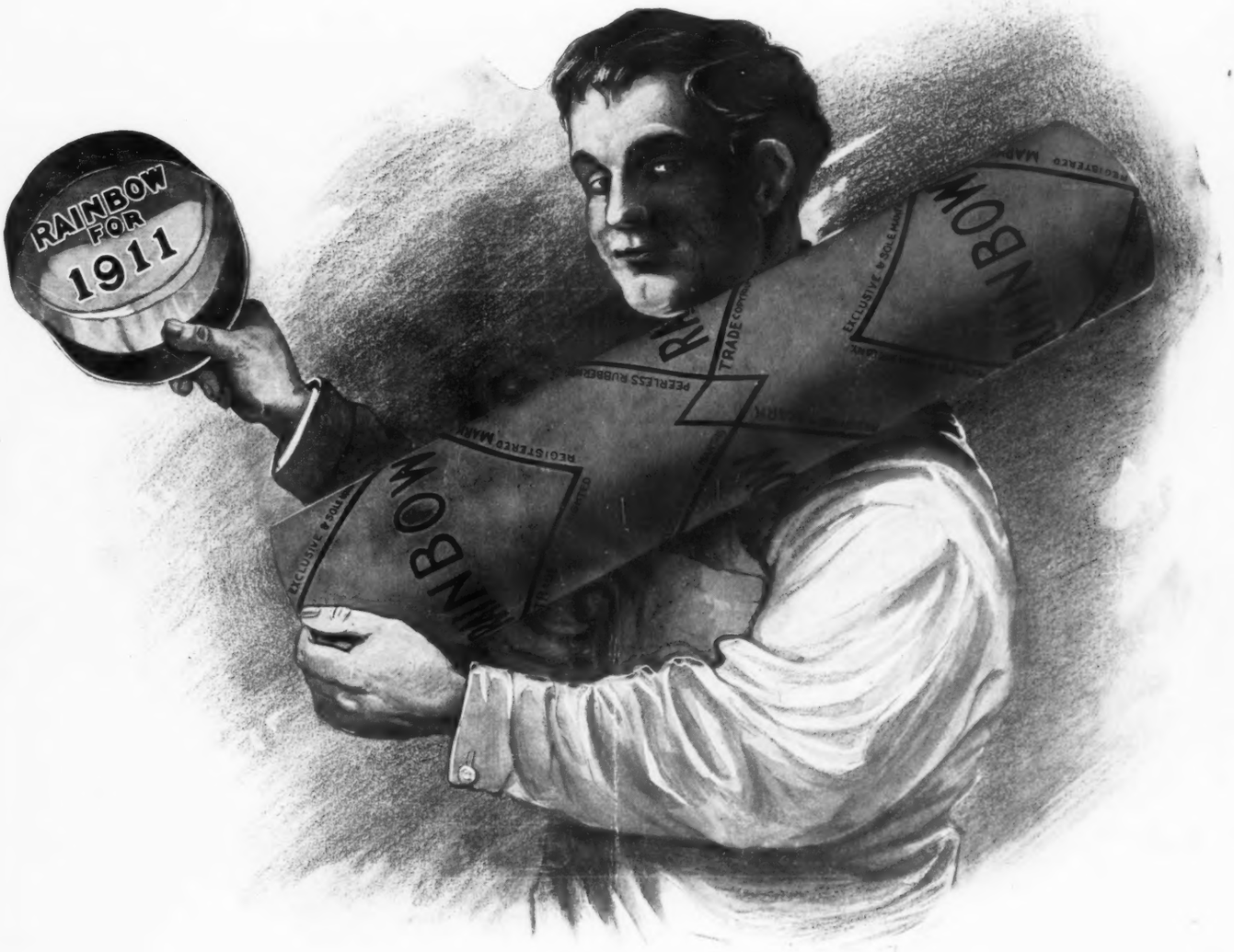

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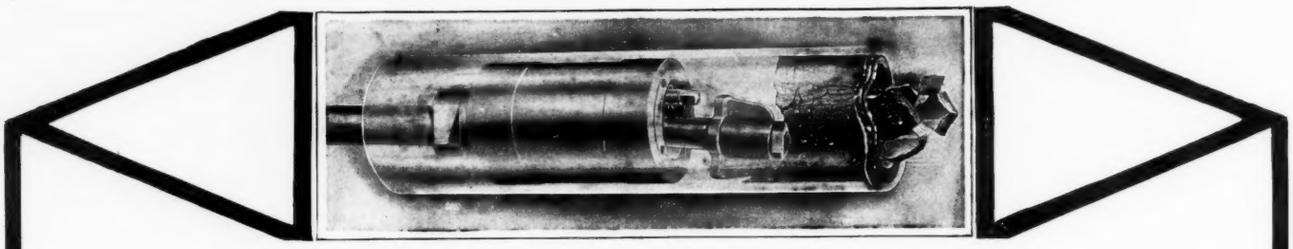

The Dean Removing Scale from the Tube of a Water Tube Boiler.

Truth About The Dean

What does a competitor say when he wants to knock the Dean?

Does he say it won't remove scale?

Does he say it isn't durable?

Does he say nobody uses it?

Does he say, "Try me device in competition with the Dean?"

Not on your life!

He's too shrewd for that.

Instead, he tells you it injures the tubes.

By playing on your gullibility he hopes to prejudice you against the Dean and make you blind to its merits as a scale remover.

The Dean is too well known to need defense, but there are those who might mistake our indifference to competitors' claims for inability to answer them.

So, we feel you ought to know the facts contained in the excerpts from the following letters:

Eddy Paper Co., Three Rivers, Mich.:

"We were afraid of the Dean at first as all boiler compound people insisted that it would loosen the tubes and play havoc with the boiler. We find, however, that the cleaner does not hurt the boiler in any way and it is remarkable the amount of scale you can get out of a boiler by its use."

A. J. West Lumber Co., Aberdeen, Wash.:

"Regarding the Dean Cleaner we bought from you some six months ago will say that we are very well pleased with it. We hesitated a long time before buying this tool as we heard a lot of talk against it from men who were selling boiler compound."

Deemer Manufacturing Co., Deemer, Miss.:

"We were informed before using the Dean that it would enlarge the flues, so in case we should want to take any of them out, it would be impossible to do so without injuring the flue sheet, also that it would open seams in the flues that would cause them to leak. We have found that there is nothing in it and the cleaner carefully used will do the work without injury to the flues."

We are particularly anxious to send you a Dean for trial in one of your boilers, if you happen to be a Doubting Thomas, for the Dean's most enthusiastic friends today were the Doubting Thomases of yesterday.

The manager of a great corporation who admitted that he was a skeptic at first writes, "The Dean will be a surprise and a satisfaction to anyone who tries it."

Shall we send you a Dean on trial?

The Wm. B. Pierce Company

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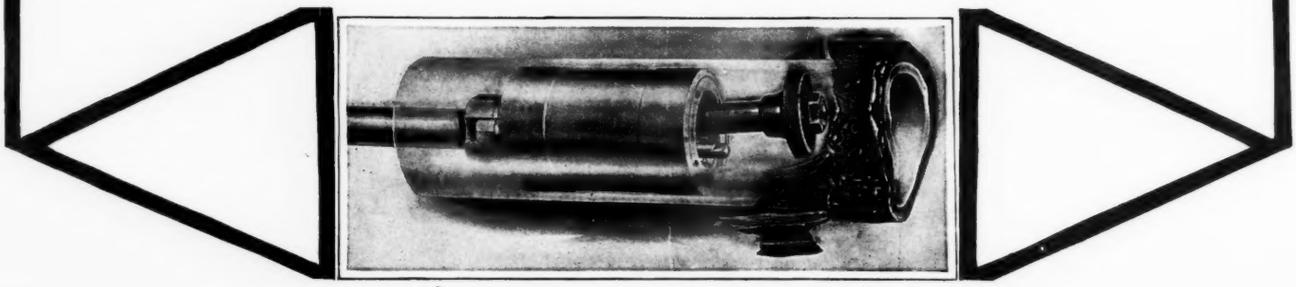


The Dean Removing Scale from the Tube of a Return Tubular Boiler.

POWER

NEW YORK, JANUARY 3, 1911

WE HAVE just passed a period of uplift, so to speak. Perhaps each of us who has been bowling merrily along, doing those things which he ought not to have done and leaving undone those things which he ought to have done, pulled up for a deep breath two mornings ago and solemnly swore, "Never again!"

This is all right and quite natural, for we have just put behind us another milestone; we are just one page nearer to the end of the chapter.

Changing the numerals from 1910 to 1911 drove home a realization of lost opportunities which we meant to improve, but which, somehow, we did not.

And so, with rustling ostentation, we "turn a new leaf," with the high resolve to "make up for lost time," to "brace up" and to do numerous other things plainly beneficial to ourselves. The doing of all these things probably amounts to nothing more than our simple duty; yet we feel uplifted, chastened, better men.

If the case is a usual one, this higher-thought thing wears off in the course of a few days, and the erstwhile better man is indulging in the same old sins of commissions and omissions.

This is mainly the result of force of habit.

This paper is made up of a consolidation of "Power" "The Engineer" "The Engineer's Review" "Science and Industry" "Steam" and "Steam Engineering"

Good resolutions are good resolutions—nothing more.

And of what use are good resolutions except to be broken?

* * *

Man is the victim of habit.

Think it over!

Does habit decide that three meals a day is the proper "caper"—then man eats three meals straightway.

Does habit decide that man shall wear his hair short—then man wears his hair short forthwith.

After a habit has been handed down for a generation or two it is given the pleasanter name of custom. The two habits just mentioned have borne the name custom so long that the bare thought of departing from either seems almost a sacrilege.

Study is a habit; neglect of duty is a habit. "Cussin'" is a habit, and so is surliness.

Forget about those impossible good resolutions and start sincerely to work on your habits. Start *today*, continue *tomorrow* and keep the work up *throughout the year.*

Break up a bad habit and institute a good one. Stick to this policy, and at the end of the current year you will find it hard indeed to draft a set of good resolutions.

Let all your habits be good ones!

Setting Horizontal Tubular Boilers

By S. F. Jeter

Directions for constructing the setting of a horizontal tubular boiler, iogether with the dimensions of different parts of the setting for various sizes of boiler, and the number of bricks required.

The usual method of purchasing a horizontal tubular boiler is to pay a stipulated price for the boiler, front, attachments, etc., delivered f.o.b. cars on the nearest railroad siding to the boiler room. From this point the purchaser usually makes a contract for the removal of the boiler to the plant and setting it in brickwork, or else turns the job over to his engineer to superintend, and hires a mason by the day to do the necessary brickwork. It frequently happens that in the small plant, which may be located in an out of the way place, that skilled masons cannot be procured, and even where brick masons are plentiful it is frequently difficult to procure one skilled in furnace work. A mason may be first class on general building work, without being able to lay up furnace work that will stand. Although it is impossible to give exact directions for doing such work, the principal features and requirements necessary to secure a good and lasting setting may be pointed out.

Foundations

The first thing that is necessary to secure a setting that will remain tight and free from cracking is a good foundation, which should be prepared before the arrival of the boiler. The manufacturer of the boiler should furnish a setting plan which will give the dimensions of the setting walls, and from this the proper dimensions and location of the foundation walls may be obtained. If a setting plan is not furnished, correct dimensions may be obtained from Table 1, used in connection with Fig. 1. In using these dimensions for a boiler already built, particular attention should be paid to the note, relative to the variations in the several dimensions as may be required, due to the hight of front, length of grate bars or the dimensions of the covering for the rear connection that are furnished, and which may not conform to the sizes given in the table. Also for flush fronts, on account of the depth of the extension sheet varying from that given for *P* in the tables, several dimensions on such settings would require modification of the figures given. The dimension *Q* for flush fronts is given uniformally two inches greater than *P* so that ample protection from the furnace heat may be afforded the extension sheet. It is impossible to give exact directions regarding the depth of foundations or the width of footings necessary, as these points depend entirely upon the nature of the soil at each plant. Formerly, stonework was used almost exclusively for building foundations, but now concrete is in general use. Where the soil is very bad and capable of supporting only light loads, a bed of concrete, properly reinforced and extending entirely over the space occupied by the setting, makes a very satisfactory foundation. It should be remembered that in arranging a foundation for boilers supported on columns, that the load on the portions of the foundations beneath the columns is more concentrated than in the case of lug-supported boilers resting directly on the brickwork, and it is necessary that additional width to the footings be provided at the base of the columns. It cannot be too strongly emphasized that the foundation must be capable of holding the boiler and setting practically rigid, for no matter how well the brickwork is set above it, a weak

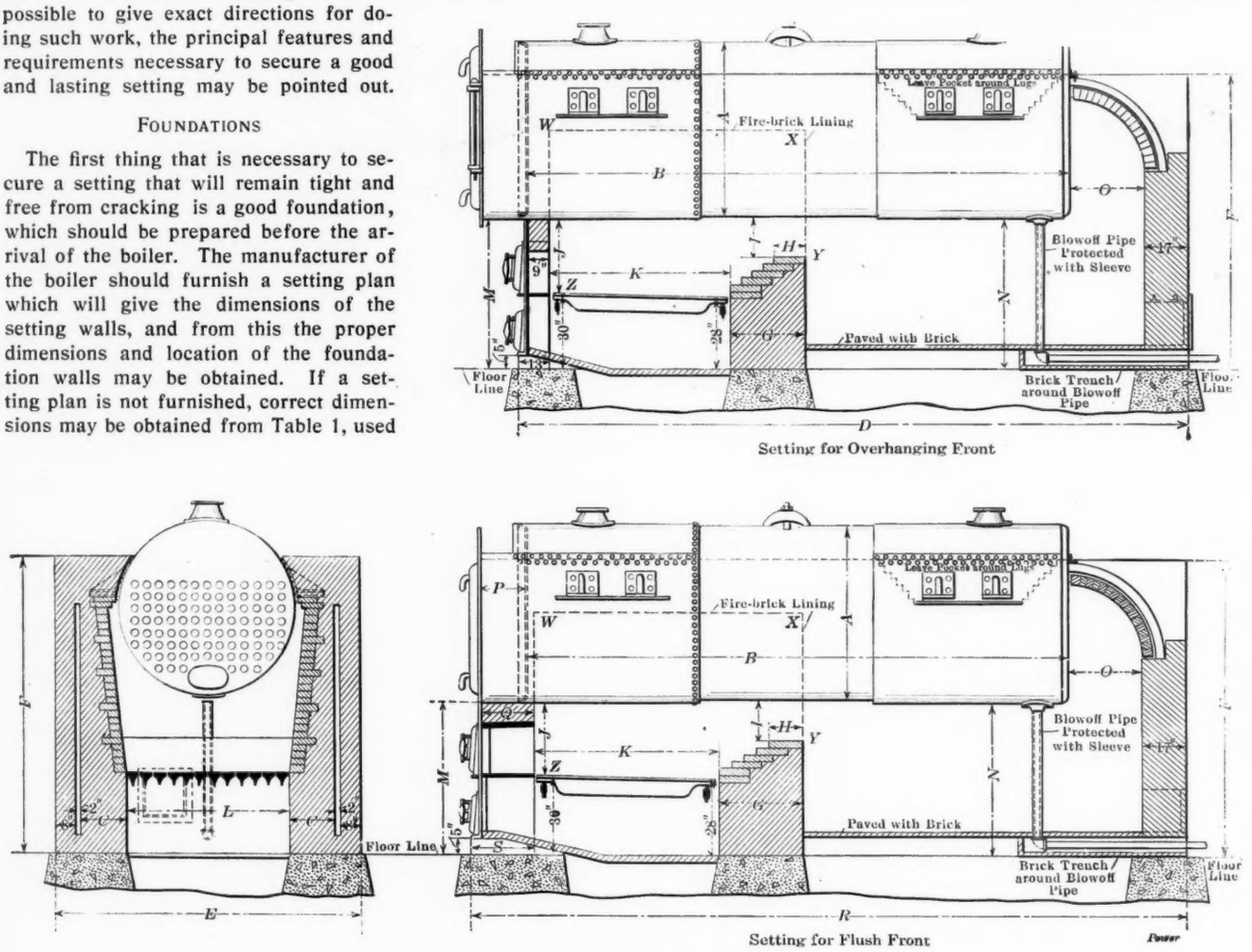


Fig. 1. Diagram of Setting for Use with Table 1

TABLE 1. DIMENSIONS FOR BOILER SETTING.

For Both Flush and Overhanging Fronts																				For Overhanging Fronts Only		For Flush Fronts Only			
Horsepower on a Basis of 10 Square Feet of Heating Surface per Horsepower			Diameter of boiler in inches.	Length of tubes, feet.	Number of tubes of each diameter.	Thickness of inside furnace walls, inches.	Over-all width of setting in feet and inches.	Hight from floor line to top of setting wall, feet and inches.	Thickness of bridge wall, inches.	Width of top of bridgewall, inches.	Top of bridgewall to shell of boiler, inches.	Top of grates at front to under side of shell, inches.	Length of furnace, inches.	Width of furnace, inches.	Floor line to bottom of shell at front, inches.	Floor line to bottom of shell at rear inches.	Distance between tube sheet and rear wall, inches.	Approximate weight of boiler and water with allowance for piping and brick covering on top of shell and flue, pounds.	Approximate ratio between grate and heating surface when three-inch tubes are used.	Number of firebricks required to set one boiler.	Number of red bricks required to set one boiler.	Over-all length of setting walls at floor line, feet and inches.	Depth of smoke box, inches.	Thickness of front wall, inches.	Over-all length of setting walls at floor line, feet and inches.	Thickness of front wall at floor line, inches.
3-inch Tubes.	3½-inch Tubes.	4-inch Tubes.	A	B		C	E	F	G	H	I	J	K	L	M	N	O					D	P	Q	R	S
53	48	44	48	12	50–3″ 38–3½″ 30–4″	17	8′–0″	8′– 0″	21½	9	10	26	48	42	56	55	22	16,000	1 to 35	800	13,500	15′–11″	15	17	16′–10″	21
61	56	51	48	14		17	8′–0″	8′– 0″	21½	9	10	26	54	42	56	55	22	18,000	1 to 35	850	15,000	17′–11″	15	17	18′–10″	21
72	63	51	54	14	60–3″ 44–3½″ 30–4″	17	8′–6″	8′– 6″	21½	9	12	28	54	48	58	57	24	23,000	1 to 35	950	16,500	18′– 1″	16	18	19′– 1″	22
83	72	58	54	16		17	8′–6″	8′– 6″	21½	9	12	28	54	48	58	57	24	25,500	1 to 40	950	18,000	20′– 1″	16	18	21′– 1″	22
99	88	86	60	16	72–3″ 54–3½″ 46–4″	17	9′–0″	8′– 8″	26	9	12	28	60	54	58	57	26	32,500	1 to 40	1,100	18,500	20′– 3″	16	18	21′– 3″	22
110	99	96	60	18		17	9′–0″	8′– 8″	26	9	12	28	66	54	58	57	26	35,500	1 to 40	1,150	20,000	22′– 3″	16	18	23′– 3″	22
130	118	108	66	16	98–3″ 74–3½″ 60–4″	17	9′–6″	9′– 6″	26	14	12	30	72	60	60	59	26	38,000	1 to 40	1,350	20,000	20′– 3″	17	19	21′– 4″	23
147	132	121	66	18		17	9′–6″	9′–6″	26	14	12	30	78	60	60	59	26	41,500	1 to 40	1,400	22,000	22′– 3″	17	19	23′– 4″	23
163	149	131	72	16	124–3″ 96–3½″ 74–4″	21½	10′–9′	9′–10″	30½	14	12	30	72	66	60	59	30	47,500	1 to 45	1,550	25,000	20′– 7″	18	20	21′– 9″	24
183	167	147	72	18		21½	10′–9″	9′–10″	30½	14	12	30	84	66	60	59	30	51,000	1 to 45	1,600	27,500	22′–7″	18	20	23′– 9″	24
203	185	164	72	20		21½	10′–9″	9′–10″	30½	14	12	30	90	66	60	59	30	56,000	1 to 45	1,650	29,500	24′– 7″	18	20	25′– 9″	24
188	175	161	78	16	144–3″ 114–3½″ 92–4″	21½	11′–3″	10′– 6″	30½	14	12	34	72	72	64	63	30	50,000	1 to 50	1,700	27,000	20′– 7″	20	22	21′–11″	26
211	197	181	78	18		21½	11′–3″	10′– 6″	30½	14	12	34	78	72	64	63	30	54,500	1 to 50	1,750	29,500	22′– 7″	20	22	23′–11″	26
234	218	201	78	20		21½	11′–3″	10′– 6″	30½	14	12	34	84	72	64	63	30	59,000	1 to 50	1,800	31,500	24′– 7″	20	22	25′–11″	26
260	242	221	84	18	180–3″ 142–3½″ 114–4″	21½	12′–3″	11′– 0″	30½	14	14	34	84	84	64	63	30	62,500	1 to 50	2,000	31,000	22′– 7″	20	22	23′–11″	26
288	268	245	84	20		21½	12′–3″	11′– 0″	30½	14	14	34	90	84	64	63	30	68,000	1 to 50	2,050	33,500	24′– 7″	20	22	25′–11″	26
310	284	267	90	18	210–3″ 168–3½″ 140–4″	21½	12′–9″	11′– 1″	35	14	14	36	84	90	66	65	32	65,000	1 to 55	2,100	32,500	22′– 9″	22	24	24′– 3″	28
334	315	297	90	20		21½	12′–9″	11′– 1″	35	14	14	36	90	90	66	65	32	70,000	1 to 55	2,250	35,000	24′– 9″	22	24	26′– 3″	28
353	320	302	96	18	248–3″ 190–3½″ 158–4″	21½	13′–3″	11′– 8″	35	14	14	36	84	96	66	65	32	70,000	1 to 55	2,350	33,500	22′– 9″	22	24	24′– 3″	28
392	354	336	96	20		21½	13′–3″	11′– 8″	35	14	14	36	90	96	66	65	32	76,000	1 to 60	2,400	36,000	24′– 9″	22	24	26′– 3″	28

NOTE—In using this table for setting boilers, particular attention must be paid to the fact that dimensions *F*, *J*, *M* and *N* must conform to the dimensions of the front furnished with the boiler and dimensions *K* and *L* with the grates furnished. On flush fronts, dimensions *Q*, *R* and *S* depend on dimension *P*, and these dimensions must be changed to conform to any changes that may be necessary in *P*. The column giving the number of common brick required refers to the overhanging-front style of setting, and for flush fronts there should be added 500 to 2500 brick, depending upon the size of boiler.

foundation will cause the walls to crack and also may cause stresses on the pipe connections to the boiler that are apt to result in a serious accident.

Unloading

When the boiler arrives at its destination it should be carefully unloaded and transported to the site of erection. In handling a boiler, one should remember that it is usually made up of a number of plates riveted together and that the tightness of each tube depends upon two expanded joints; therefore, the boiler cannot be handled as if it was a chunk of pig iron. The writer has seen a boiler deliberately dropped from the side of a flat car, and the mechanic superintending the job expressed surprise that a block on which it happened to land had dented the shell. The nozzles are most likely to be damaged in handling; and pipes or bars should never be stuck in the tubes to aid in moving the boiler.

Placing the Boiler in Position

It is best to place a boiler in the correct position with the front in place before commencing the brickwork; if the boiler is to be supported on lugs resting on the brickwork it should be placed about a half inch higher than the desired final position, to allow for lowering on the brickwork when the supports are removed. When a boiler is to be hung from beams it can be placed in the correct position at once. None of the weight should be carried by the boiler front, and to insure against this ½ to ¾ inch clearance should be left between the bottom of the shell and the front. Ample clearance between the front and shell is especially important in the lug-supported type in order to allow for settling.

The front end of a boiler should be placed about 1 inch higher than the rear to aid draining through the blowoff pipe when washing out; this also allows an extra inch depth of water over the rear tube ends, which is a precaution against damage from low water. To level a boiler crosswise it is necessary to consider two points, the top line of tubes and the faces of the steam nozzles. Every boiler manufacturer endeavors to have the face of the steam nozzle parallel to a line across the tops of the tubes; but owing to the fact that the nozzle is finished in a lathe and then riveted to the boiler shell, the surface of the flange is sometimes out of true with the top line of tubes. Usually slight differences of this kind can be taken up in the packing of the joint, but if the top line of tubes and the face of the nozzle are out enough to prevent a proper joint being made, the boiler should be set so that the tubes are level crosswise and a special flange used to fit the nozzle to bring the main steam pipe vertical. Many engineers view the matter from a piping standpoint alone and endeavor to level the boiler by the face of the steam nozzle; this, however, is not correct, because the short length of surface at the top of the steam nozzle precludes accurate leveling from this point, and also because it is of more importance that the tops of the tubes be level than the flange of the nozzle. The angularity of the nozzle face can be remedied by the use of a special mating flange, but the tops of the tubes across the boiler not being level means a higher water line and consequently a reduction of steam space which cannot be remedied.

Blocking or barrels placed beneath the shell are generally used to hold a lug-supported boiler in position while the setting walls are being built; however, barrels are preferable to blocking, as they are less in the way of the brick masons. Two heavy oil barrels in good condition can be depended upon to support a 66-inch by 16-foot boiler, if the blocking below them and on top is arranged so that the load is distributed evenly over all the staves. Additional barrels should be used for larger boilers and the blocking on the top arranged so that the load will be distributed evenly between the barrels. If good barrels are not available, a cribwork of blocks placed under the front and rear ends of the shell will serve the purpose. In placing such supports care should be used in the arrangement of the blocking so that it will not interfere with the building of the setting walls.

Materials Required

Some masons still use common lime mortar in building boiler settings, but a much better and more lasting job can be obtained by adding cement to the bonding mixture. First, regular lime mortar is made, using three-quarters of a cubic yard of good, sharp sand to one barrel of lime. After this has been made up in the usual manner, a mixture of sand and cement is made, using two barrels of sand to one barrel of cement (four bags of cement); this mixture of sand and cement is added to the lime mortar and it is then ready for use. This quantity of material should make enough mortar to lay about one thousand brick. If all the mortar cannot be used at once, the sand and cement mixture should only be added to such portion of the lime mortar as will be required for immediate use, as it is difficult to keep it in proper condition for use over night after the cement has been added. Fire clay is the only bonding material that should be used in laying the firebrick and for this purpose it should be mixed with water to about the consistency of buttermilk, so that the bricks may be dipped in it and rubbed together when laying them. About two barrels of fire clay are required to lay one thousand brick.

The temperatures attained in the furnaces of return-tubular boilers are generally moderate, and it does not require a specially high grade of firebrick to withstand the heat; but there is more need of mechanical strength to withstand the wear incidental to the rubbing of the fire tools and breaking off clinkers. On this account a medium grade of firebrick, costing about $22 to $25 per thousand, will be generally found most suitable. Firebrick that are made especially with a view to resisting the very high temperatures are usually mechanically weak and soft and they are also the most costly. For arches in dutch ovens, where there is no danger of hitting the brick with the fire tools, the higher grade of brick generally gives the best service. The common brick used for setting should be well burned and selected for strength rather than beauty.

To estimate the number of common brick required for a boiler setting, figure the number of cubic feet of wall that is to be laid with this kind of brick and multiply by 23; the result will be the number of brick required. In making calculations for the number of brick, no deductions should be made for openings in the setting walls for cleaning doors, etc.; for the waste from breakage and cutting will require all of the extra brick figured in this way. Where fire lining is laid 4½ inches thick and with every sixth course a header, eight firebrick should be figured for each square foot of wall surface lined in this manner. If the lining is to be 9 inches thick and with every sixth course tied to the common brick with a header, fifteen brick should be figured for every square foot of wall surface lined.

TABLE 2. WALL THICKNESSES.

Walls All Common Brick, Inches.	Common Brick Wall Lined with Firebrick on One Side, Inches.	Common Brick Lined with Firebrick on Both Sides, Inches.
8½	13½	18½
12¾	17¾	23
17	22¼	27¼
21¼	26½	31¾
26	31	36
30½		40½
35		45
		49½

Thickness of Wall

Draftsmen usually specify a nominal thickness for the walls on a setting; and often the brick mason (who does not know how much change may be made without affecting the work) is troubled in endeavoring to meet the given dimensions without cutting the brick. For standard-sized brick, Table 2 gives about the proper wall thicknesses to specify, so that they may be laid without cutting the brick.

The sizes of common brick vary slightly with each locality; but the standard is 8¼x4x2 inches and the standard size for firebrick 9x4½x2½ inches. Although

the standard sizes of firebrick are so different from the common brick, they lie together correctly because the firebrick are laid brick to brick, while the common brick have about ½ inch of mortar between them.

Design of Setting Walls

Return-tubular boilers are usually set with an air-spaced wall, as illustrated in Fig. 1. Many claims are made as to the benefits derived from such construction, one of the chief being that it lessens the radiation losses by keeping down the temperature of the exposed wall surface. The air space does reduce the temperature of the outer wall surface, but introduces other losses that probably outweigh the gain in economy due to this feature, and it is very doubtful if this form of construction, from an economical standpoint, is better than a solid wall.

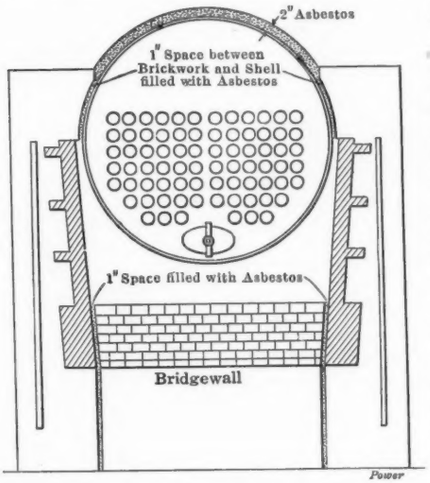


Fig. 2. Showing Clearance at Ends of Bridgewall and Around Shell

The chief advantage of the air-space construction is that when properly built it tends to prevent the cracking of the outer wall surface and, therefore, makes a better looking setting. One very important point in the design of setting walls, to prevent cracking, is the method used to join the ends of the bridgewall with the side walls. Usually a mason will build the two at the same time, and tie the bridgewall rigidly to the side walls. This method of construction is almost certain to result in cracked side walls, because the bridgewall expands when heated and pushes out the side walls. With the wall having an air space this does not necessarily show on the outer wall unless the two are tied together at this point.

There are two ways of preventing trouble from the expansion of the bridgewall. One is to leave the ends of the bridgewall about an inch away from the side walls, as shown in Fig. 2, packing the space with asbestos or mineral wool. The elasticity of the packing allows for the expansion of the bridgewall and it prevents the space from becoming clogged with ash and cinders. The other way to accomplish the same purpose is to build a recess about 4½ inches deep in the side walls having the same shape as a vertical section of the bridgewall, and build the ends of the bridgewall into this recess, leaving 1½ inches of clearance at each end for expansion; this method of construction is shown in Fig. 3.

There are many different ideas regarding the proper shape to be given to bridgewalls and the correct distance that should be left between the top of the wall and the shell of the boiler. The chief function of a bridgewall is to limit the length of the grate surface by presenting a barrier beyond which the spreading of the fuel is prevented; it also aids in mingling the unburned gases and air, so as to cause complete combustion before reaching the tubes. The exact shape or hight of the bridgewall does not greatly affect the attainment of these functions, but it may play an important part in tending to cause or prevent trouble at the girth seams due to overheating. Where girth seams are located in the vicinity of the bridgewall, the top of the wall should be so shaped and of such a distance below the shell that the products of combustion will not impinge directly against the seam. A distance of at least 10 or 12 inches should be left between the top of the bridgewall and the shell to prevent overheating of the sheets, even in the absence of seams; and the top of the bridgewall should be built straight across and not follow the contour of the shell as is sometimes done. All the bricks on top of the bridgewall should be laid as headers, as illustrated in Figs. 1 and 3, so that they may be better able to resist being dislodged by the fire tools. It is much better to rake the top of the bridgewall as shown in Figs. 1 and 3, instead of trying to cut the brick to make them present a smooth slope; for when the bricks are cut, clinker adheres more tenaciously to the cut surfaces than it does to the original surface of the brick.

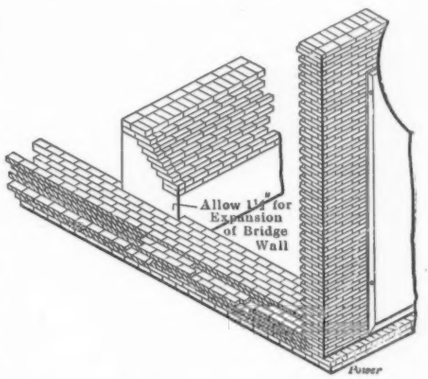


Fig. 3. Method of Allowing for Expansion of Bridgewall

The side walls of a boiler are generally battered as shown in Fig. 1; and this is good construction, especially for a lug-supported boiler; for in making repairs to the lining on the side walls of the furnace the firebrick may be removed and still leave ample support for the boiler.

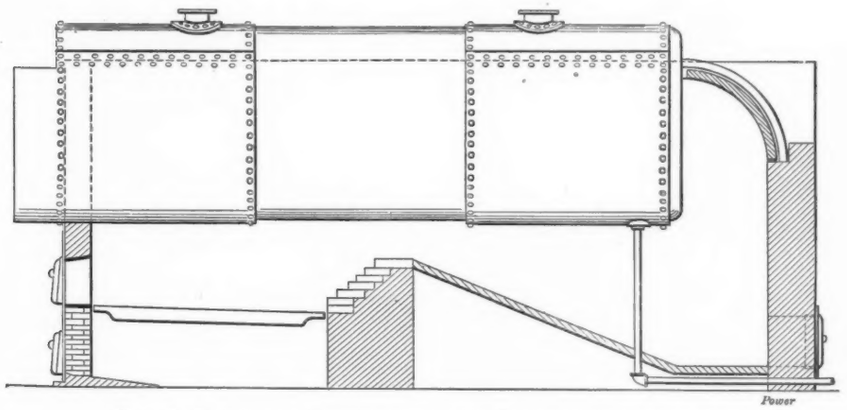


Fig. 4. A Common Arrangement of Combustion Chamber

Combustion Chamber

The combustion chamber at the rear of the bridgewall is a very important feature of the setting, tending to aid complete combustion, especially if bituminous coal is used. The rear edge of the bridgewall should be built vertical, and the space behind it down to about the level of the floor should be left open as in Fig. 1, and not filled up and paved as in Fig. 4, which is common practice. The deep combustion chamber at the rear of the bridgewall tends to cause a whirl in the air and gases coming over it and greatly aids in their proper mixture. It also affords storage capacity for the fine ash and cinder that is carried beyond the bridgewall. The practice of filling the space behind the bridgewall to conform to the contour of the shell, as is sometimes done, cannot be too strongly condemned, for it seriously interferes with the accessibility for inspection of the most important surfaces of the boiler, and is certain to prevent complete combustion, if bituminous coal is

used. Convenience in cleaning out the combustion chamber is obtained by arranging the bottom of this chamber as illustrated in Fig. 1; so that the blowoff pipe passes out below the paving, and the cleanout door, which is usually located in the rear wall, is placed on a level with the paving so that no obstacle is offered to raking out the ashes. The blowoff pipe should be placed in a brick trough, the bricks on top being arranged so that they may be readily removed for inspection. This arrangement also admits of the blowoff pipe being placed above the boiler-room floor without interfering with free access to the cleanout doors. The vertical section of the blowoff pipe should be protected from the direct impingement of the flames by slipping a pipe sleeve over it; or a form of protection which is equally as good, with the blowoff pipe accessible for inspection, may be made by laying loose firebrick in front of the pipe in the form of a V.

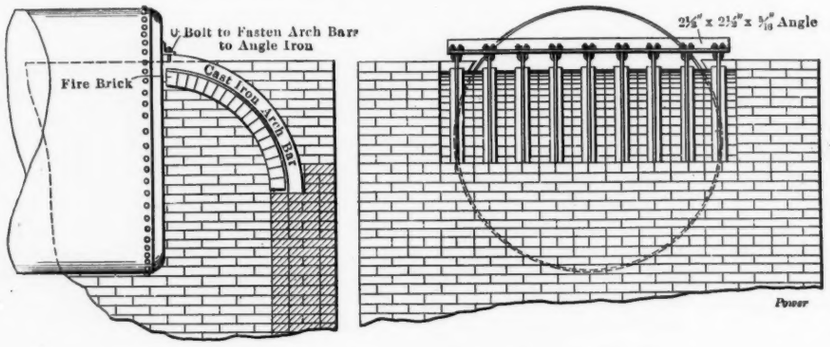


FIG. 5. BEST FORM OF COVERING FOR REAR CONNECTION

FIREBRICK LINING

The amount of wall surface that is required to be lined with firebrick is largely a matter of opinion; some engineers prefer to line all of the inner surfaces that are swept by flame and heated gases; but, although this makes a good and lasting setting, it adds considerably to the cost. If the front wall and the side walls over the space indicated by the letters *W X Y Z*, Fig. 1, are lined, together with the bridgewall, and the balance of the setting is laid with good, hard, burned red brick, a satisfactory and very durable job will result. Every fifth or sixth course of firebrick should be a header course to properly bind the lining to the main wall. In laying fire lining too much emphasis cannot be put on the necessity of using the minimum amount of bonding material. Fire clay, which is the only kind of material that should be used for this purpose, should be mixed with water to the consistency of buttermilk and the bricks dipped in it and rubbed down on each other as they are laid. When too much fire clay is used between the bricks where exposed to high temperatures, the clay will fuse and apparently acts as a flux to run out the brick material, resulting in wearing away of the bricks at the joints; a condition that may be noted with improperly laid linings.

BINDER BARS

Although it has been the general custom to place binder bars on side walls of settings, it is a debatable question as to whether they are of any real benefit or not, except possibly near the front and rear ends of the setting. When a boiler is set with a dutch oven, there is absolute need of binder bars or their equivalent to carry the thrust of the arch, but no such need exists with the ordinary return-tubular setting where the boiler is hung, and probably not where the boiler is supported by lugs resting on the setting walls.

ALLOWANCE FOR EXPANSION

An important point upon which depends the prevention of cracks in the walls of the setting, is the proper provision for expansion of the boiler. In supporting the boiler on lugs it is generally attempted to secure this feature, in part, by providing rollers under one pair of lugs (usually the rear lugs), as shown in Fig. 1. These rollers prevent a lengthwise thrust on the walls due to the expansion of the shell; but it is doubtful if they are of much real value because they do not provide for any movement across the setting. For instance, in a 72-inch by 16-foot boiler. the longitudinal distance between the centers of the lugs is about 8 feet, while the distance between centers across the boiler is about 7 feet; hence, the movement across the setting that should be cared for is about as great as it is lengthwise, and the rollers do not aid the movement in this direction. The method of making allowance for expansion between the shell and setting is shown in Fig. 2, where a 1-inch space is left between them and the space filled with plastic asbestos or asbestos rope. The brickwork should not be allowed to touch the boiler at any point, and special care must be taken to keep it free from the rear supporting lugs, pockets usually being left in the walls for this purpose. Another point where clearance is of vital importance is around the pipe connections to the water column and the blowoff pipe, for, unless proper freedom is allowed at these points, there is danger of the pipes being broken off.

BACK CONNECTION COVERING

This is one of the most difficult points about a boiler setting to keep tight. There are numerous methods of arranging the covering at this point, and one of the best ways to accomplish this, which is in common use in the West and South, is illustrated in Fig. 5. The usual arrangement of this form of covering is to have an angle-iron strap bolted to the boiler head, and the ends of the arch bars rest on the leg that extends outward; but owing to the fact that the angle is exposed to the direct heat of the gases it burns off in a short time. A better arrangement is to fasten the angle to the tops of the arch bars by means of U-bolts, so that they will all line up together. If desired the angle iron may be bolted to the boiler head, although this is not necessary. With this form of covering the arches follow the movement of the boiler head; and by covering the whole surface with plastic asbestos about 2½ inches thick, a tight job is insured. One of the desirable features of this form of covering for the back connection is that it presents a straight line across the head above the tubes, affording ample protection against overheating to the portion of the head above the water line, without interfering with the free passage of heated gases to any of the tubes.

Another method of closing in the back connection that is commonly used throughout the East, is illustrated in Fig. 6. In setting this type of arch, care must be used that the head above the water line is not exposed; and it is sometimes necessary to partially block off one or two of the outside tubes to accomplish this. In the arrangement of all types of covering for the back connection the fusible plug must be left uncovered so that it is freely exposed to the products of combustion.

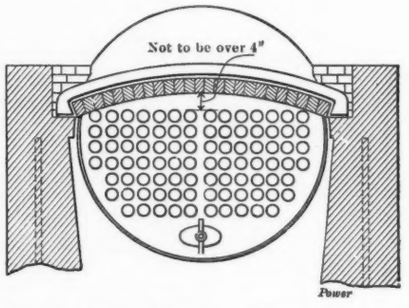


FIG. 6. CROSS ARCH FOR COVERING BACK CONNECTION

Boilers Supported on Columns

Where boilers are hung from beams or channels supported on columns and more than one boiler is used, a column is often placed in the dividing wall between boilers; where this is the case too great care cannot be exercised to keep such columns from being overheated. In such cases there should be at least 13 inches of brickwork between the column and the fire and a 2-inch air space around the column, with free ventilation in this space. To accomplish this, air should be admitted near the bottom of the column through an open duct not less than 10 inches square. These requirements, where a column 8 inches in diameter is used, mean that the minimum wall thickness between the boilers at the grate level must be 38 inches.

Covering the Top of the Boiler

The best covering for the exposed surface on top of a boiler, and the one that will reduce the radiation losses to a minimum, is 85 per cent. magnesia from 2 to 3 inches thick, the outer layer being made with a hard finishing cement. A cheaper covering, but one that will resist wear better than the magnesia and still reduce the radiation losses to a low point, is made of asbestos, but it should be of good grade. The usual covering consists of a layer of bricks laid on edge; but such covering only has cheapness and durability to recommend it, as it is practically worthless as an insulator.

Cost of Setting

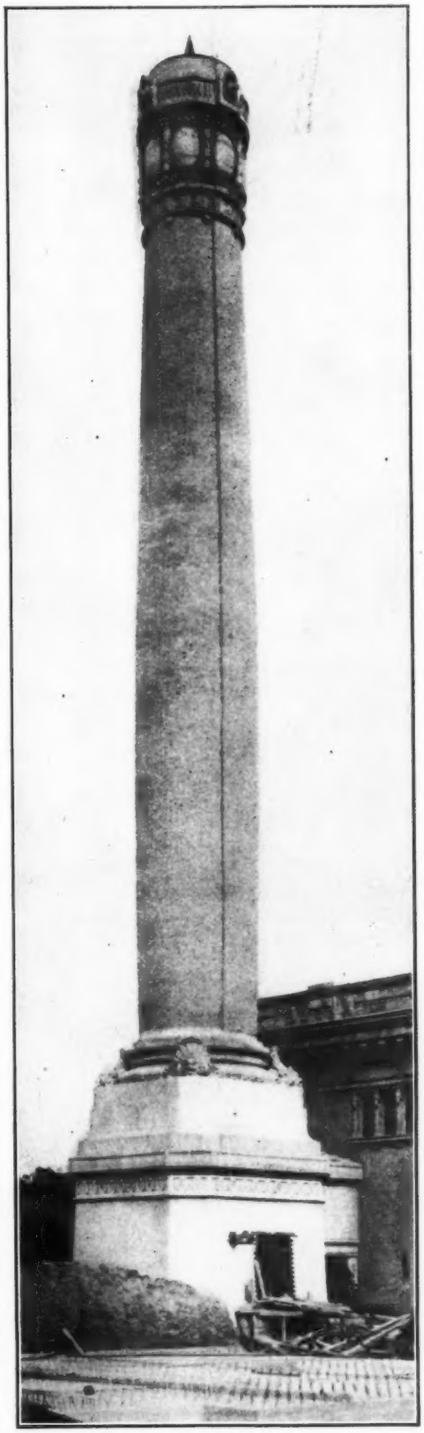
With common bricks at $9 per thousand and firebricks from $22 to $25 per thousand, mason's wages at 60 cents per hour and laborers at 30 cents per hour, a very accurate estimate of the cost of a boiler setting may be obtained by figuring $22 per thousand, laid, for the common brick and $45 per thousand, laid, for the firebrick. The cost, laid, will rarely exceed $25 and $50 per thousand for common and firebrick respectively.

Setting Water Columns

In setting a boiler attention should be given to the proper location of the water column; it should be placed so that the lowest gage cock is at least 3 inches above the tops of the tubes, and the lowest point of vision in the gage glass at least ½ inch above the tops of the tubes. The latter point is of great importance, for it is often a temptation for an engineer to take chances as long as he can see water in the glass; although he may realize that it is lower than safety demands and it is therefore best to prevent the water level from being seen after the lowest point for safety has been reached.

Starting Boilers

No matter how carefully a boiler setting has been built, it can be badly damaged and cracked, by carelessness in starting up. No setting which has just been completed should be operated in regular service without a thorough drying out; this drying is accomplished by keeping a slow fire under the boiler for at least a week, and longer if circumstances will permit. The interior of a new boiler should always be thoroughly cleaned of oil, dirt, etc., and the best way to accomplish this is to boil slowly with a strong solution of soda ash, and in this way the drying of the setting walls and the cleaning of the boiler can be accomplished simultaneously.



Chimney at Queen Lane Filter Plant, Philadelphia

A Handsome Chimney

In these days when the high cost of living is a problem which baffles satisfactory solution, the industrial-plant proprietor is less than ever inclined to spend good hard cash to satisfy the artistic tastes of the members of the community where the plant is located. A chimney is bought for the amount of draft it will create and gases it will handle. The purchaser usually wants the most stack for the least money. Consequently, few chimneys are on view that have any work on them for purely ornamental purposes.

Occasionally, however, conditions may be such that the expenditure of money to produce ornamental effects in chimney and power-house design is justified. Such a set of conditions exists at the Queen Lane Filter Plant, Queen Lane and Fox street, Philadelphia. The accompanying figure is a view of the chimney at this plant. It is reputed to be the handsomest chimney in America. The plant is located in a high-class residental section of the city and any building or plant which would tend to detract from the desirability of the neighborhood would, of course, have a bad influence on the value of the surrounding property. For this reason it was found justifiable to go in for the esthetic in the matter of building ornamentation.

The chimney is 125 feet high and 6 feet in diameter inside. It serves six horizontal return-tubular boilers, the total rated capacity of which is 900 horsepower. The lower portion or base of the chimney is of granite and terra-cotta. The upper portion or shaft is constructed of radial brick, faced with pearl-gray blocks to match the terra-cotta ornaments of the cap and base. The round column or shaft of the chimney has a double entasis; that is, a double curvature, which is used to overcome the optical illusion which a straight shaft gives of being concaved. This is the design which was applied by the Greeks and Romans to the columns in their temples.

The general scheme of ornamentation is Italian Renaissance. The architect employed conventionalized water forms as motives of decoration, both on the chimney and the power house itself, as being indicative of the character and use of the structure. It will be noticed in the view that use is made of shells, cat tails, dolphin heads, star-fish, etc., cleverly placed, as motives of decoration instead of the more commonplace garlands and flowers, etc.

In designing the ornamentations of the cap, it was necessary to give them very

bold and vigorous relief, otherwise they would have been lost to view on account of the great hight; hence, the large medallions or disks, surmounted by the huge lion heads, which form so conspicuous a climax to a structure unique in the annals of chimney building.

The total cost of the chimney complete was $15,500. Under the existing circumstances it is undoubtedly true that the expenditure of this sum was justified. A radial-brick chimney of the same capacity for an industrial plant could be purchased complete for approximately $4000.

The chimney was designed by Architect William E. Groben, of Philadelphia, and erected by the M. W. Kellogg Company, contractors, New York City.

The Old Burden Water Wheel

One of the most interesting landmarks of Troy, N. Y., is the old Burden waterwheel. Its days of usefulness, however, are over and it is fast going to decay. The old mills for which it furnished power are in ruins and all that remains of the once busy industry are the toppled down walls, the old waterwheel and the iron penstock leading to it from a grass- and weed-covered canal. The new Burden iron works are located near the Hudson river and point to the march of progress in the steel industry; the old discarded mill indicates the primitive generation of power sixty years ago.

Old overshot wheel built at Troy, N. Y., in 1851, said to be the largest in the world. It is 60 feet in diameter, 22 feet wide and at two revolutions per minute developed 600 horsepower.

Henry Burden, the founder of the original Burden iron works, was the inventor of many appliances, but his greatest achievement was in designing the immense waterwheel, shown in Fig. 1, which was constructed in 1851. It is of the overshot type and was capable of developing 1200 horsepower. It is 60 feet in diameter and 22 feet wide and contains 36 buckets, each 6 feet deep. These are shown in Figs. 2 and 3.

FIG. 1. OLD BURDEN WATERWHEEL BUILT IN 1851

The axis is composed of six hollow cast-iron tubes keyed into flanges, from which diverge two hundred and sixty-four 2-inch iron rods terminating at the outer edge of the wheel.

The two axis flanges are made with bearing shafts 12 inches in diameter. The bearings in which these shafts rested are each supported by an iron frame which set on a brick foundation built between the two upright brick piers shown in Figs. 1 and 4.

Power was transmitted to jack shafts by means of small gears meshing into a toothed rim placed on the outer circumference and outside edges of the waterwheel as shown in Figs. 1 and 4. The jack shafts transmitted the power to the mill rolls by means of shaftings which extended from the flywheel and gears as shown at the left of Fig. 1.

The flow of water was governed by a rod and handwheel, the upper end of the extension rod supporting a worm which meshed with a gear, mounted on a shaft that extended from one side of the wheel to the other, on top of the flume, and supported by suitable bearings, as shown in Fig. 4.

By turning the handwheel and rod, the shaft and gear were revolved in the direction desired, and by means of gears and racks, the latter being attached to the stem of each gate in the flue, the gates were opened or shut. The water was thus regulated in flowing from the penstock to the four outlets over the buckets placed between the three metal distance pieces.

The water was brought to the wheel through an iron penstock, which extended out over the waterwheel, the water coming from the canal through the farther gate shown in Fig. 5. The second gate was for the purpose of emptying the canal.

This old waterwheel is said to be the largest in the world. When running at a speed of two revolutions a minute, between 500 and 600 horsepower was developed. A wheel of larger diameter was constructed at one time at Wales, but being of less width and depth of buckets developed less power.

The First Steam Cylinder Used in America

In a glass case in the National Museum at Washington is preserved the cylinder of the first steam engine ever run in America. The following extracts from a letter of the Hon. Joseph T. Bradley, Associate Justice of the Supreme Court of the United States, dated at Washington, September 20, 1875, to David M. Meeker, of Newark, N. J., into whose possession, we understand, the cylinder had come, contains pretty nearly all that is known in reference to this interesting relic:

FIG. 2. WITH THE BUCKETS FULL OF WATER THE WHEEL REVOLVED TOWARD THE FOREGROUND

"The steam engine of which this is a

portion of the cylinder was the first ever erected on this continent. It was imported from England in the year 1753 by Col. John Schuyler for the purpose of pumping the water from his copper mine opposite Belleville, near Newark, N. J. The mine was rich in ore, but had been worked as deep as hand or horse power could clear it of water.

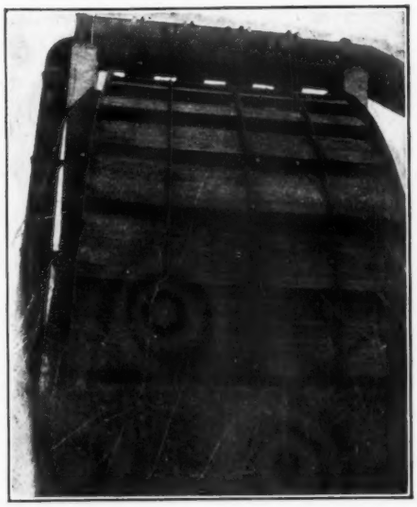


FIG. 3. SHOWING DESIGN OF BUCKETS

"Colonel Schuyler having heard of the success with which steam engines—then called fire engines—were used in the mines of Cornwall, determined to have one in his mine, and accordingly requested his London correspondent to procure an engine and to send out with it an engineer capable of putting it up and in operation. This was done in the year named and Josiah Hornblower, a young man then in his twenty-fifth year, was sent out to superintend it.

"Mr. Hornblower's father, whose name was Joseph, had been engaged in the business of constructing engines in Cornwall from their first introduction in the mines there—about 1740—and had been an engineer and engine builder from the first use of steam engines in the art—about 1720.

"The engines constructed by him and his sons were the kind known as Newcomen engines, or Cornish engines. That brought to America by Josiah was of this construction. Watt had not then invented his separate condenser, nor the use of high pressure, but it is generally said that for pumping purpose the Cornish engine has still no superior.

"About 1760 the Cornish mine was worked for several years by Mr. Hornblower himself. The approach of the war in 1775 caused its operation to cease. Work was resumed, however, in 1792 and was carried on for several years by successive parties. It finally ceased altogether in this century and the old engine was broken up and the materials disposed of. The boiler, a large copper cylinder standing upright, 8 or 10 feet high and as large in diameter, with a flat bottom and dome-shaped top, was carried to Philadelphia. A portion of the cylinder was purchased by some person in Newark.

"In 1864 I met an old man named John Van Emburgh, then 100 years old, who had worked on the engine when it was in operation in 1792. He described it very minutely, and I doubt not accurately. It is from his description that I happen to know the kind of engine it was, although from the date of its construction and the use to which it was put there could have been little doubt upon the subject."

FIG. 5. ALL THAT IS LEFT OF THE OLD GATE CONTROL

FIG. 4. PENSTOCK AND ARRANGEMENT OF GATE CONTROL

The pumps for drawing water from the lake at Chicago to dilute the sewage of the city are to be operated by electric motors. Recently, one of the triple-expansion vertical engines was replaced by a 750-horsepower motor. This motor, though but a small machine compared to the steam engine, has the same pumping capacity, namely, 40,000 cubic feet per minute. It is operated with current supplied by the hydroelectric transmission system of the sanitary district.

Gravity Turbine Oiling System

By Hugh Hughes

Description of the design and operation of a gravity oiling system for a vertical turbine. Bypasses are extensively used so as to secure flexibility of feed and continuity of operation.

Much has been written about engine-oiling systems, and much may be written on turbine-oiling systems. The simplest form of the latter is the gravity feed which in all respects resembles that employed for engines. Its possibilities, however, when applied to turbines are not fully appreciated. By arranging a few bypasses, with valves properly placed, almost any combination of feed may be obtained. Referring to the left half of the accompanying figure, *A* is the lower receiving tank; *B* the suction to oil pumps; *C* the discharge to the upper receiving tank *D*; *E* the feed to the reservoir *F*; *G* are sight feeds on the line *H* to top bearing and the line *J* to middle bearing; *K* and *L* are the returns from the top and middle bearings respectively. The greatest head of oil that that can be brought upon the upper bearing with this system is about 3 feet.

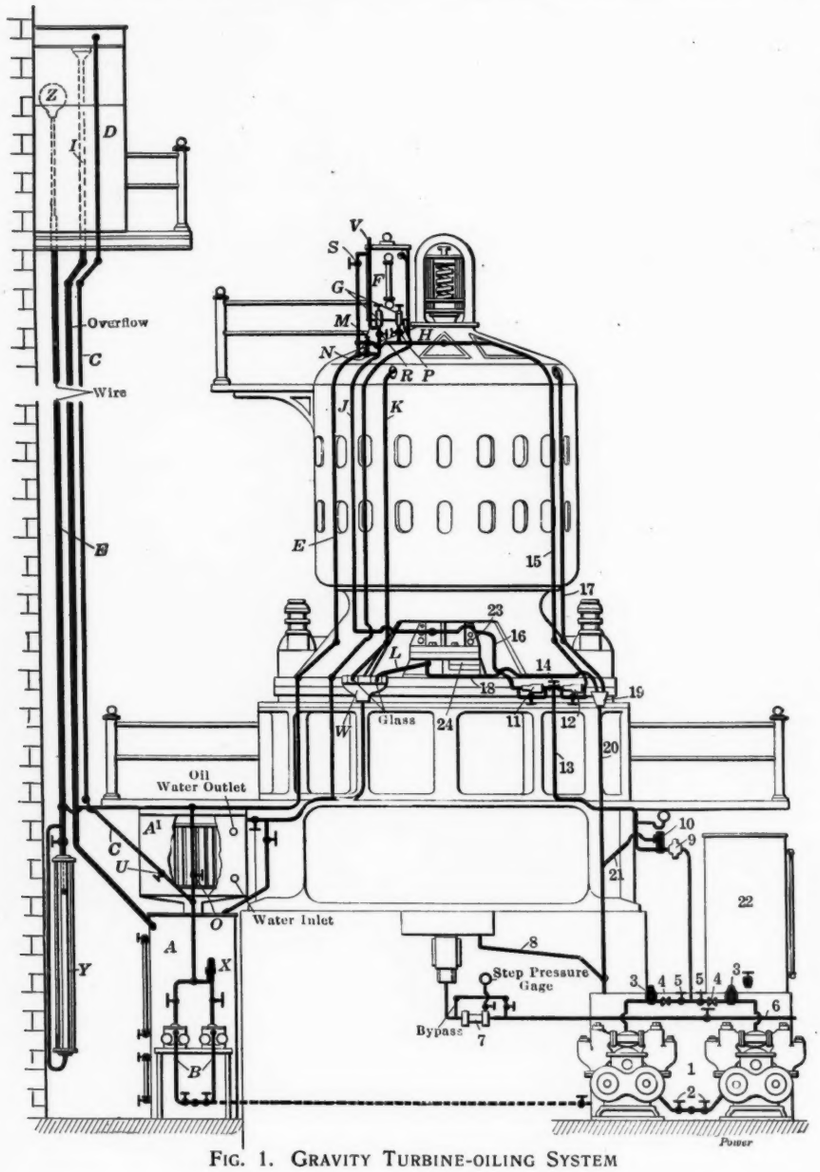


FIG. 1. GRAVITY TURBINE-OILING SYSTEM

If the bearing has no tendency to heat, this is quite sufficient; but it costs but very little more to pipe in the bypass *M*, by means of which the direct pressure of upper tank *D* may be brought on this bearing. The connection *N* is a bypass that will bring this pressure on the middle bearing also. If still more pressure is required, the oil pump may be made to discharge directly into the feed line through the bypass *O*.

In practice, the bypasses would be employed somewhat as follows:

To feed more oil to the top bearing, close the valve *P* and open valve *M*; to middle bearing, close the valve *R* and open *N*; to both bearings, simultaneously close the valves *P*, *R* and *S* and open *M* and *N*. If, for any reason, the upper tank *D* and the reservoir *F* are allowed to run dry, open the valve *O* and close *U*, and pump directly into the feed line, for it will take some time for the oil to reach the reservoir if allowed to follow its usual course, especially if air enters the pipe. In this dilemma, if the bypass *O* is not included in the system, draw a bucketful of oil from the tank *A*, or any other source, and empty it into the reservoir *F*; keep doing this until oil in sufficient volume is delivered to it from the top tank *D*.

Some believe that better lubrication is obtained by piping an air vent below the sight feeds as shown at *V*. Others, in cases where the flow of oil is subject to frequent stoppages, prefer to keep the reservoir *F* for emergencies only, the oil being led to a tee above the sight feeds and the reservoir outlet closed by a valve which is opened only when the regular feed stops. The returns from the bearings should be piped separately, and at some point should be open, so that the amount of oil passing through each bearing can be seen and its temperature ascertained. Some have the two returns discharge into a funnel open to the atmosphere as shown at 19. If the engine room is dusty, especially if it is subject to coal dust, it is better to have the returns discharge through oil-cup glasses with tin or brass covers as shown at *W*. It is good practice where the pump discharge is piped directly to the feed line, as through the bypass *O*, to mount a small relief valve as shown at *X*. Very often it is the case that the oil line in the glass gage of the upper tank cannot be seen from the floor below. An arrangement for indicating the quantity of oil in the tank is shown at *Y*. This is simply a glass tube piped to a continuation of the downward feed pipe. The ball *Z* floats on the surface of the oil in the tank *D*, and is guided by a short length of quarter-inch tubing. To the end of this is attached a straight piece of light wire of a length sufficient to reach the glass gage *Y*. Any dark object pendant at

the end of the wire is very easily distinguished if the glass is filled with clear water. The internal piece *I* of the overflow pipe from tank *D* is screwed hand tight only, so that it can easily be removed when it becomes necessary to drain the tank. It will be noticed that the oil pumps are shown with the oil ends outward. This way of installing is preferable to the universal practice of side exposing, for one pump is as accessible for repairs as the other. With

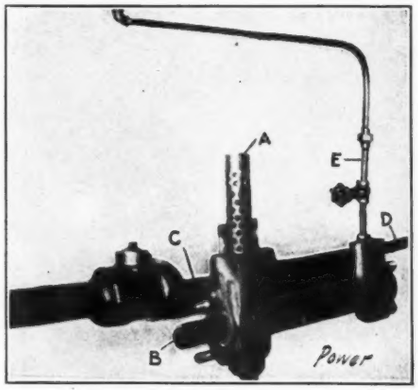


FIG. 2. THE BAFFLER

the other arrangement, if repairs are needed to the inside pump, one must lean over the outside pump, which may be in service, and the repair man is lucky to escape without burns.

In many turbine plants oil coolers are now being installed, so that the oil is used not only to lubricate a bearing but also to cool it by carrying away the heat generated. One form of cooler is shown at A^1. The water is obtained by tapping the discharge of the circulating pump. Cooling the oil condenses the vapor it may contain, which falls to the bottom of the receiving tank *A*. This tank should therefore be equipped with two separate sight glasses as shown, to establish clearly the true quantity of water present. When only one long glass is used the water at the bottom will be forced up in it, by the weight of the accumulating oil. I have seen a barrel of oil turned into the sewer by an attendant, who mistook the water level in the glass as indicating the water level in the tank. So far, it has been assumed that the step bearing is subject to hydraulic pressure. When oil is the balancing medium employed, it is good practice to use the same system of oiling the upper bearings as described above. The only change necessary is to substitute the larger storage tank 1 for the receiving tank *A*. The course of the oil to the step bearing would be from storage tank 1 through suction valves 2 to the step-bearing pump; thence under relief valve 3, through the check valve 4 and the stop valve 5 to the accumulator 6 and the baffler 7.

The oil returns from the step and guide bearings to the storage tank through the pipe 8. Valves 3, 4 and 5 are shown elevated in the figure merely to obtain space to map them. In some plants oil for the upper bearings is taken directly from the step-bearing pumps discharge, the pressure being lowered by the reducing valve 9. Tanks *A* and *D* and the pair of small oil pumps on the left-hand side are now done away with. If the top reservoir *F* is retained, the gravity feed is again restored on the two upper bearings; if it be removed and the feed led directly to the bearings then the relief valve 10, and the two bafflers 11 and 12, one on each feed branch, should be installed. The course of the oil now is: From the step-bearing pumps discharge through the pipe 13, to the cross valve 14, where it divides and enters the baffler 12 and the pipe 15 for the top bearing, and the baffler 11 and pipe 16 for the middle bearing. The returns 17 and 18, from the top and middle bearings respectively, enter the funnel 19, from which to the storage tank they have pipe 20 in common. The relief-valve discharge is at 21.

With such a system it is in line with good practice to have a reserve tank 22 to be used for sudden emergencies or to replenish storage tank automatically. In some plants where the turbine is run continually, and much vapor is present in the oil, the storage tank is in duplicate and each is used alternately, thus allowing time for the oil to cool and the water to settle and be drawn out from one while the other is in service. The storage tank should be placed at as great a distance below the step bearing as it is possible in order to allow for some head for the oil return. Some oil step bearings are provided with a small pipe open to the atmosphere and entering the space between the oil and the carbon packing to facilitate the flow of the return oil and prevent its intermixture with the condensed steam used to seal the vacuum. This can also be somewhat aided and air prevented from entering the return by piping it on the principle of an ejector to the common return 20 as shown in the figure. The down-flowing oil in this pipe, if the connections are properly made, will have a tendency to siphon out the waste oil from the guide bearing.

Fig. 2 shows an adjustable baffler. *A* is the strainer, *B* the baffler screw, *C* is the oil inlet and *D* the oil outlet; *E* is a pressure-gage connection. At the other end of the baffler screw *B* is an adjusting screw the head of which, when the baffler is in place, rests against the cover on the outlet side of the baffler chamber. The more this bolt is backed out of the baffler screw, the more will the oil pressure be reduced. If manipulation of this screw does not give the desired flow of oil a piece may be sawed off the end of the baffler, or one having a coarser thread may be substituted. As a rule, the baffler is designed to reduce the pressure about 25 per cent., though this depends materially on the viscosity of the oil. Altogether, it is an ingenious contrivance for causing trouble if not regularly inspected.

Most readers have seen in POWER sectional cuts of the upper and lower bearings of a Curtis turbine, and some have conceived the idea that they are very complicated affairs. Fig. 3 shows these members for a 2000-kilowatt turbine when stripped of all supports. A drip pan, which is generally bolted to the bearing support, and an oil deflector cast on the turbine shaft to throw the waste oil by centrifugal force into the drip pan would about complete the bearing. The part *A* is the top bearing, which fits tightly into the top bracket of the turbine. The governor and governor hood must be removed to get at it. Oil enters the bearing through the small holes *B* and leaves around the bottom or by the row of upper holes *C*. The parts *D* are the two halves of the middle bearing, which are locked together by an outer shell, also in halves and bolted together. These locking bolts are to be seen at 23 in Fig. 1, while 24 is the drip pan mentioned. The part *E*, Fig. 3, is the shaft lining.

A description of the newer styles of bearings will be found in back numbers of POWER. If a bearing has a tendency to throw oil into the generator fields, or if steam escapes from around the high-pressure carbon packing and commingles with the oil in the drip pan of the middle bearing, felt washers held gently against the shaft by sheet-iron straps should be secured to the bottom of the oil pans or top of the bearing brackets. In some plants

FIG. 3. PARTS OF UPPER AND LOWER BEARINGS

a practice is made of oiling the high-pressure carbon rings upon starting the turbine. An oil cup, secured to a pipe which ends as close as possible to the shaft, is filled with freshly mixed graphite and cylinder oil. While the turbine is slowly revolving and more or less air is being drawn in around the carbon bearing, the mixture is let out of the cup. Oil here, however, must be used sparingly as it is liable to gum the carbon packing rings and cause them to stick in their receptacles.

Modern Steam Superheaters

By Warren O. Rogers

A general description of the few successful types of American superheater, dealing with their features of design and application to various types of boiler, flooding and control of the furnace gases passing over the members.

Less than ten years ago hundreds of engineers held the opinion that, although the water consumption of a steam engine would be reduced by superheating the steam, the extra fuel required would offset the saving in the water consumption. But when the facts are fully considered there can be no doubt in the minds of those who have taken the trouble to inform themselves that, with a properly designed plant, the superheater will add to the economy.

In 1860, when the question of superheating was first taken up, the pressure carried in steam boilers ranged from 25 to 50 pounds per square inch. With these low pressures tallow was found to answer very well for cylinder lubrication, but the use of superheated steam brought trouble to the engineer, because the high temperature of the steam dried up and decomposed this animal oil.

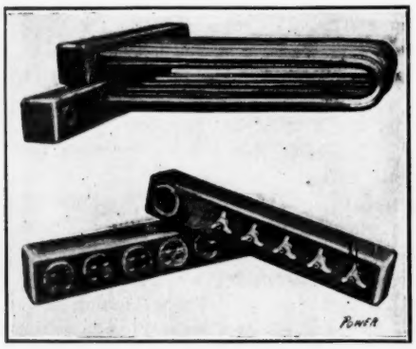


FIG. 1. HEADER AND TUBES OF PARKER SUPERHEATER

About this time, however, engine builders began building compound engines which demanded higher boiler pressures, and as this demand was met, the desirability of superheating the steam decreased. Furthermore, this increase in steam pressure made the use of superheated steam all the more difficult. For this reason superheating was practically abandoned for a period of thirty years.

About 1890 superheating was taken up by engineers in European countries, and was carried on successfully, especially by the Germans. The problem was solved by using a high-grade mineral oil for lubricating purposes, together with valves and cylinders of suitable design. The demand for superheated steam brought out the superheater, of which there are several types made in this country that give very satisfactory results.

There has been, and still is, a difference of opinion as to how high steam should be superheated. A common range is from 100 to 200 degrees of superheat, and 150 degrees is considered a maximum figure by many engineers.

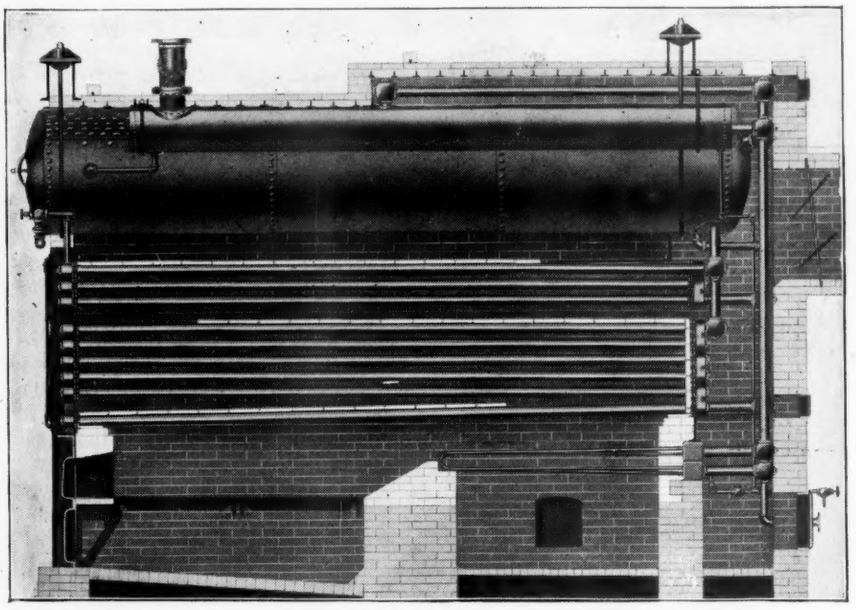


FIG. 2. PARKER SUPERHEATER AS APPLIED TO A PARKER DOWN-DRAFT BOILER

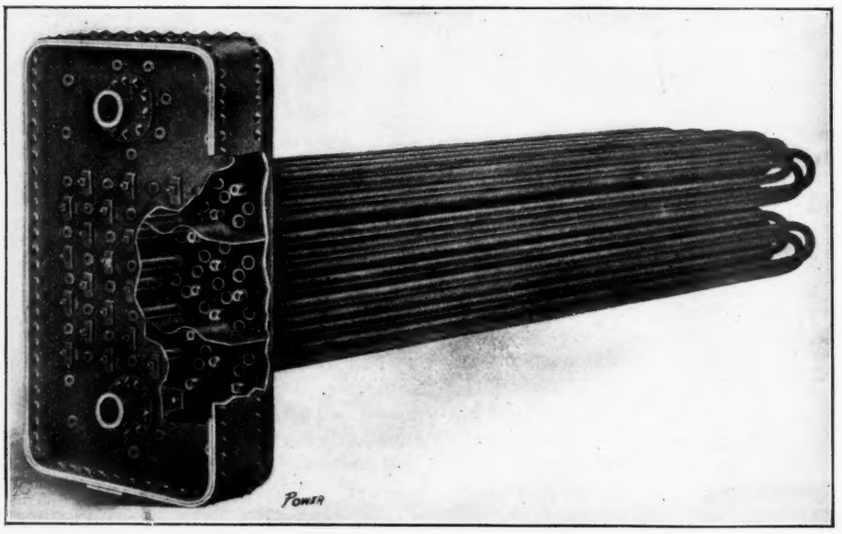


FIG. 3. DETAILS OF TUBE AND HEADER CONSTRUCTION IN HEINE SUPERHEATER

A common objection to using superheated steam is that it has been the cause of many ills which are not encountered when using saturated steam. Packing troubles, however, have been practically eliminated; lubrication of the valves and piston can be satisfactorily maintained, and troubles from failure of pipes and fittings are being greatly reduced. As to the superheater itself, one company that has made superheaters for fifteen years has never been called upon to make a single repair due to damage from excessive temperatures.

A superheater contained within the boiler setting is perhaps the most efficient type when the steam is not to be superheated more than 200 degrees. Such superheaters require no additional space in the boiler room, unless it is an in-

creased hight in the boiler setting, and the amount of piping required is very small. A superheater so arranged is, however, subject to the fluctuating temperatures of the furnace, for, if the boiler is forced, the superheater is correspondingly affected by the increased temperature of the furnace gases. It is also subjected to the cooling effects of air admitted through the fire doors, which naturally cools the gases passing to the uptake.

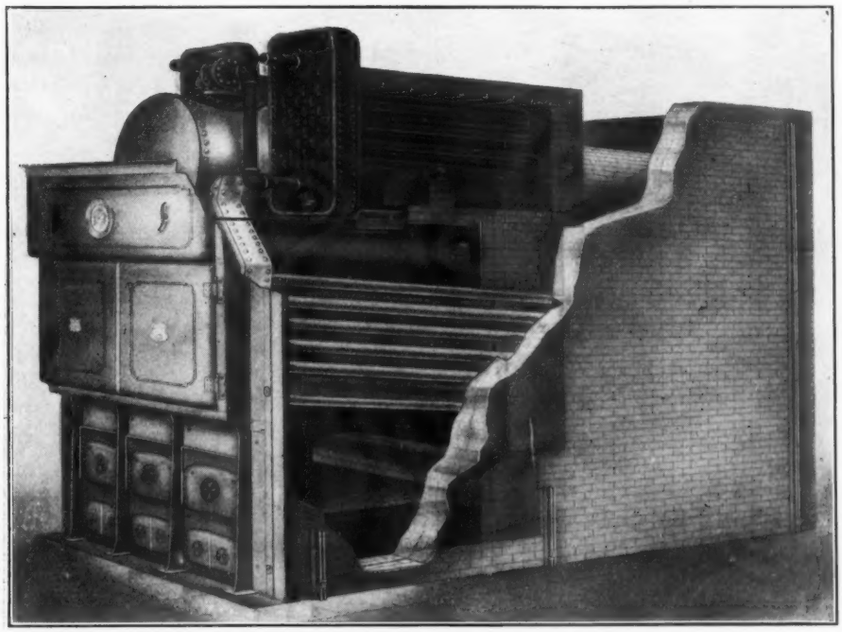


FIG. 4. HEINE BOILER AND SUPERHEATER

Cast iron, which was at one time largely used in the construction of superheaters, is gradually being replaced by stronger metals, especially where high steam pressures are concerned. At best it is a treacherous metal when subjected to temperature changes, and cannot compare in point of safety with forged steel, cast-steel and drawn-steel sections.

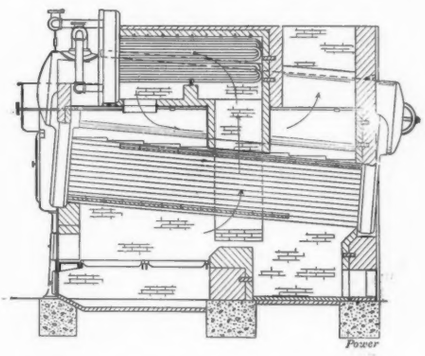


FIG. 5. SECTIONAL VIEW, SHOWING PATH OF GASES

In this article several types of superheaters will be taken up and attention given to their principles of design, methods of attaching to the boiler, flooding, and their general characteristics.

PARKER SUPERHEATER

This superheater, which is used in connection with the Parker down-flow boiler, consists of a number of seamless drawn-steel tubes of small diameter, the number and length being proportioned to the degree of superheat desired. The tubes are bent into U forms and are expanded into steel headers. The general design of the headers and U tubes is shown in Fig. 1; front and back views of the headers are also shown.

An unique feature of this superheater is the use of the superheater drum or separator between the superheater and the steam nozzle. This prevents carrying the flooding water into the steam line, and saves the heat that would be wasted by draining the superheater.

Connection is made from the lower head of the superheater to the steam space of the steam drum of the boiler by a tube expanded into malleable-iron fittings. The top superheater header is connected to the steel superheater drum in a similar manner. At the end of the drum farthest from the superheated-steam inlet is the steam outlet, from which connection to the steam main is made. This drum, which is shown in Fig. 2, acts as a separator, and has sufficient capacity to handle all of the water discharged from the superheater and connections when getting up steam in a cold boiler, or when putting a banked boiler on the line, the circulating water being returned directly to the superheater from the drum. The superheater is fitted with a drain connection for returning all excess water to the boiler drum. Thus the superheater automatically floods itself with the water of condensation when the fires are banked and requires no special attention in this regard.

Fig. 2 shows the general appearance and arrangement of this type of superheater. It is placed near the fire and thus requires only a small amount of heating surface.

All of the hot gases from the burning fuel must pass to the rear of the boiler before passing up among the boiler tubes, owing to the baffle wall placed on the upper side of the lower row of tubes for the greater portion of their lengths. This brings the superheater practically in the direct path of the hottest gases.

The superheater is located near the bottom row of boiler tubes where most

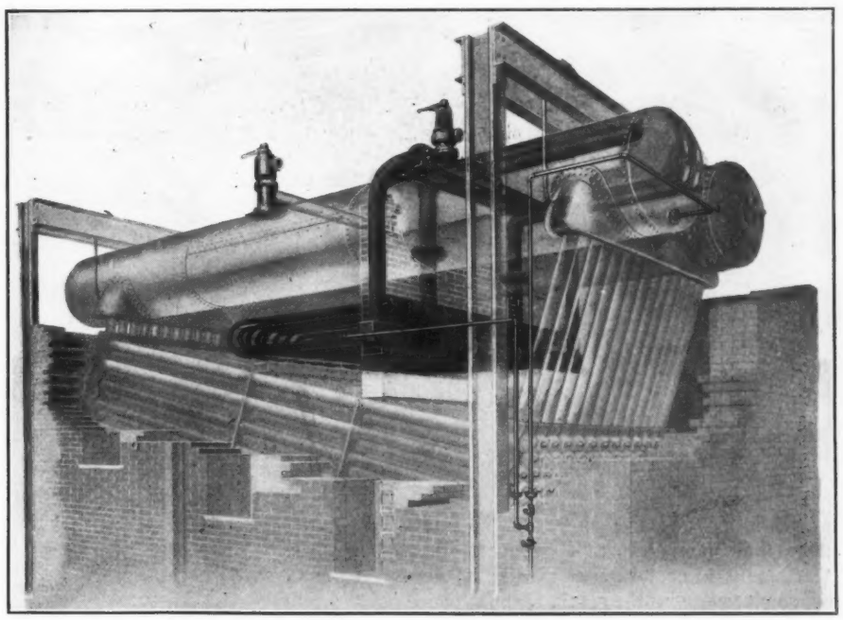


FIG. 6. PHANTOM VIEW OF BABCOCK & WILCOX SUPERHEATER ATTACHED TO A BOILER OF THE SAME MAKE

of the steam is generated, and any change in the condition of the fire affects the boiler and superheater simultaneously, maintaining a remarkably uniform degree of superheat. Another advantage of this location is that the size of the boiler setting is not increased, and there are no losses due to radiation and air leakage.

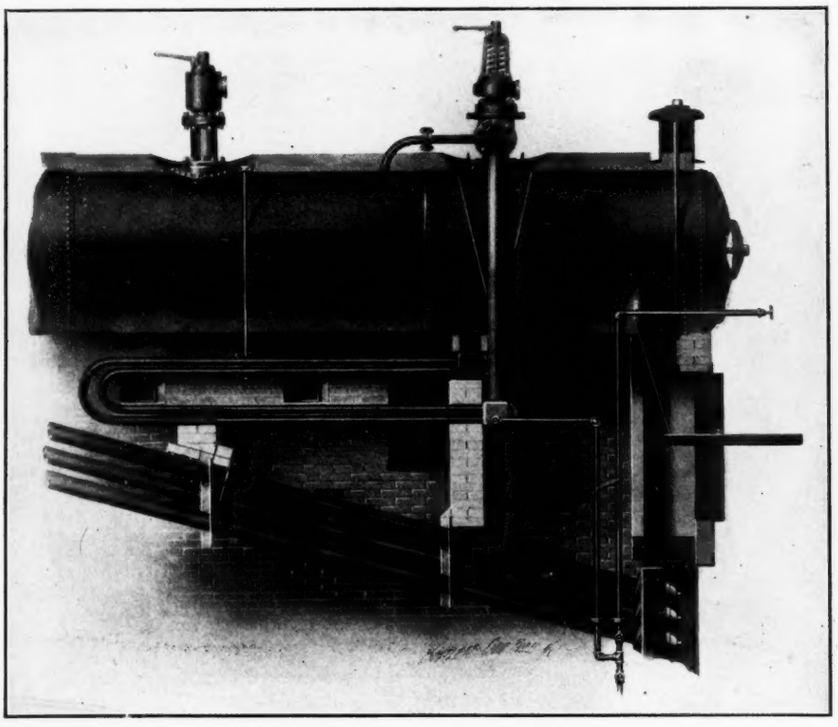


FIG. 7. SHOWING POSITION OF HEADERS AND FLOODING ARRANGEMENT

HEINE SUPERHEATER

Although the members of a superheater may be similar in construction, a difference is found in the design of the header. This is particularly noticeable in the Heine superheater, made by the Heine Safety Boiler Company.

This superheater consists of a header box in which U tubes are expanded and the flat sides of the header are strengthened by staybolts. The interior of the header box is divided into three compartments by means of sheet-iron diaphragms, as shown in Fig. 3. These diaphragms cause the steam to change its direction of travel four times in passing through the U tubes before entering the steam pipe leading to the main steam header.

This type of superheater and method of setting is shown in Fig. 4. It is located at the side of the boiler drum toward the front, and just above the last passage of the boiler gases. It is incased in brickwork which is lined with firebrick on the roof.

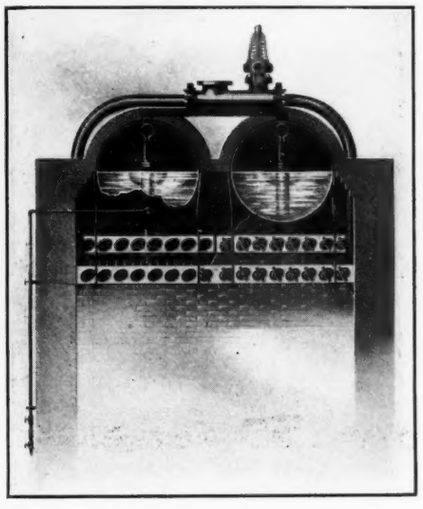


FIG. 8. END VIEW OF MANIFOLDS IN BRICKWORK

In order that the hot gases may be carried direct from the furnace to the superheater a small flue is built in the side wall of the boiler setting. In this flue the hot gases make two passes around the superheater tubes, as shown in Fig. 5. The flow of gases is controlled by a damper placed at the outlet end of the flue. When the damper is closed the circulation of the hot gases is stopped and when the heat from the gases in the flue in which the superheated is located, has been absorbed, saturated steam only is delivered to the steam main. Owing to this method of controlling the hot gases, various degrees of superheat up to the capacity of the apparatus can be obtained.

The saturated steam-outlet connection

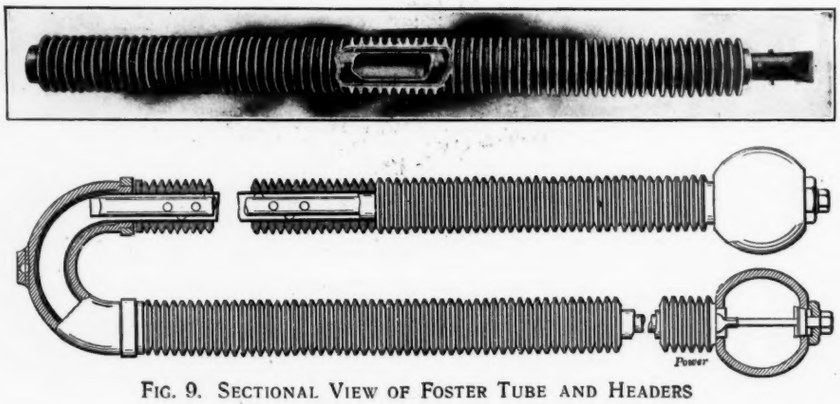


FIG. 9. SECTIONAL VIEW OF FOSTER TUBE AND HEADERS

By placing the superheater in the path of the hottest gases passing from the furnace their temperature is reduced, and this results in cooler gases passing to the economizer and up the stack. If the superheater were placed in the flue it would not reduce the temperature of the gases so much on account of the smaller difference between the temperature of the steam and the escaping gases. One of the reasons for the uniformity of superheat in this superheater is that the steam and water of the Parker boiler are separated by a diaphragm and the boiler never primes.

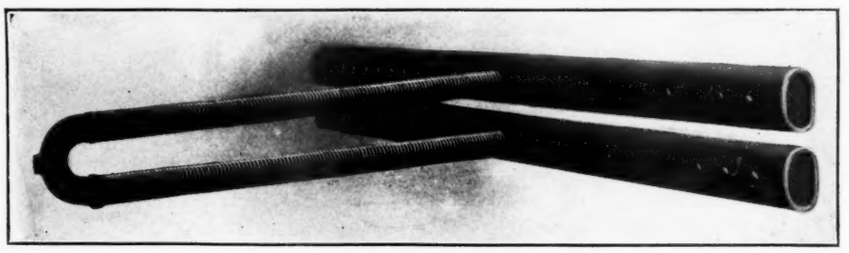


FIG. 10. EXTERIOR VIEW OF TUBE AND HEADERS

from the boiler is made to the lower end of the superheater box, and the steam, after passing through the tubes, goes to the main steam header.

Owing to the location of the superheater above the boiler, and having no connection to it below the water line, the tubes are never flooded, nor is it necessary, on account of the complete control of the gases by the damper, which is easily operated by thermostatic control.

The exterior surfaces of the superheater tubes are smooth and such accumulations of soot as stick to them do not seriously interfere with the transmission of heat, as provision has been made by which the soot is removed by means of a blower introduced through the hollow staybolts passing from one header plate to the other. By this means the tubes of the superheater are cleaned without interfering with its operation or that of the boiler.

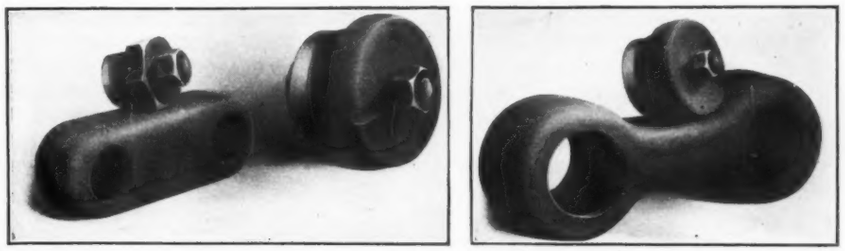


FIG. 12. HEADERS AND PLUGS

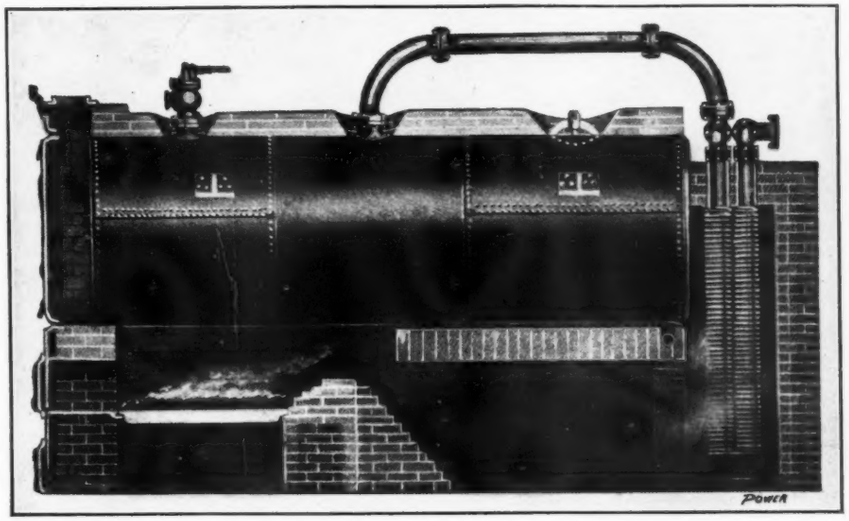


FIG. 13. FOSTER SUPERHEATER IN REAR COMBUSTION CHAMBER OF RETURN-TUBULAR BOILER

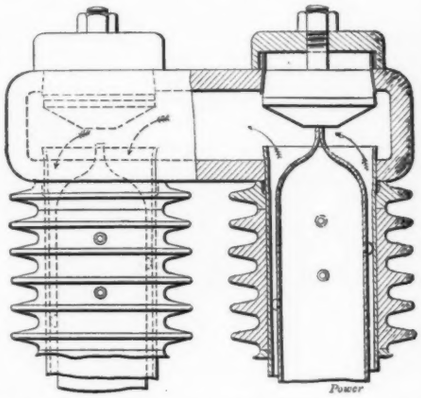


FIG. 11. SECTIONAL VIEW THROUGH RETURN HEADER

BABCOCK & WILCOX SUPERHEATER

A design of superheater that has found favor, and is extensively used, is the type made by the Babcock & Wilcox Company. It is made entirely of wrought steel and is so constructed that no joints are in the direct current of the hot gases. The superheater is made up of 2-inch No. 8 gage seamless steel tubes bent to a U form, and of square headers or manifolds of forged steel, into which the tube ends are expanded. The tubes are arranged in clusters of four, and an oval handhole opening is made in each header opposite each cluster of tube ends A finished seat is formed on the inside of the header around each handhole, and a wrought-steel handhole cap having a fixed stud with a threaded end, is used for closing each opening; the joint is made tight by a gasket. The stud of the cap passes through a forged-steel guard, and a wrought-steel nut secures the cap in place against the gasket on the inside of the header. No cast-iron parts or screwed joints are employed in the construction of this superheater.

When used in connection with the Babcock & Wilcox boiler, the superheater is placed in the triangular space below the steam and water drums and above the inclined tubes, as shown in Figs. 6 and 7. It is supported independently of the brickwork by hangers from the drums of the boiler, in a position where it is accessible for inspection and does not in any way interfere with the cleaning or repairing of the boiler.

Both manifolds are built into the brickwork of the hanging bridgewall over the second set of baffles, as shown in Fig. 8. The tube ends and handholes are thus protected, while the space back of the hanging bridgewall affords access to all handhole fittings. If necessary, the tubes can be rerolled or replaced with

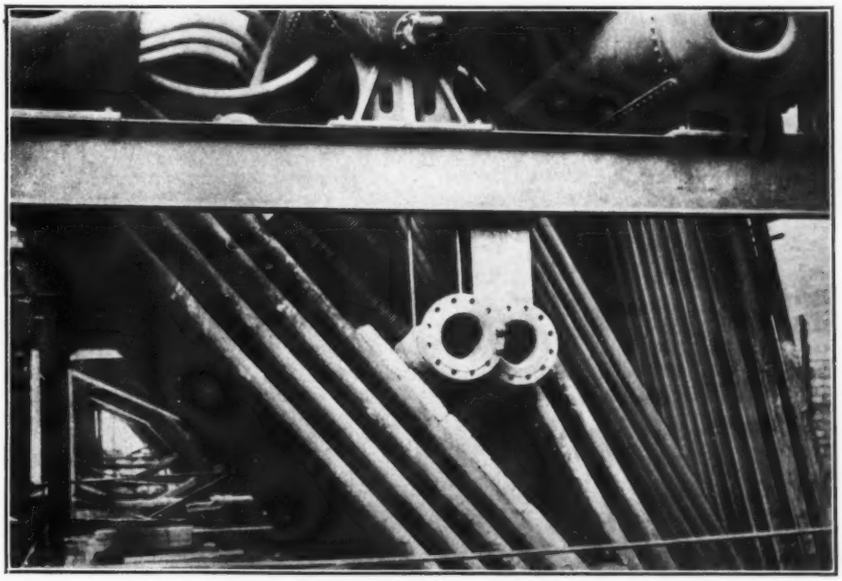


FIG. 14. SUPERHEATER ATTACHED TO STIRLING BOILER

no more trouble than would be experienced in a like operation on the boiler. The U form of the tubes and the fact that the expanded joints are out of the path of the hot gases insure against stresses is uniformly distributed through the upper manifolds and passes through the tubes to the lower manifolds. The superheater safety valve on the outlet fitting is set to open before the safety valves thus preventing excess superheat. By means of the external flooding piping the superheater can be flooded when steam is being raised on the boiler, thus protecting it from any possible overheating during this period.

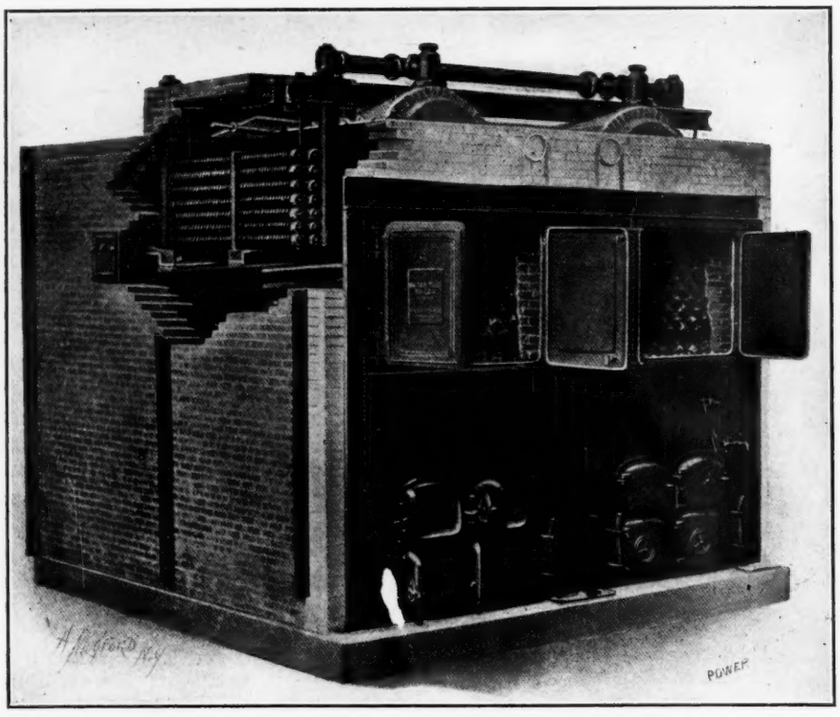


FIG. 15. FRANKLIN BOILER AND FOSTER SUPERHEATER

Owing to the position the superheater occupies in the setting an even flow of heated gases over the heating surface is insured, and at the same time the area of the gas passage is not reduced so as to affect the operation of the boiler. As the external surfaces consist only of smooth, seamless tubes, there are no recesses in which ashes and soot may collect to any considerable extent, and at any time the tubes can be thoroughly cleaned with a steam jet from a lance passed through the dusting doors in the side walls of the setting.

The entire heating surface of the front pass of the boiler is between the superheater and the furnace, and wide variations of temperature are avoided in the chamber in which the superheater is located, such fluctuations of superheat that do occur being relatively unimportant.

FOSTER SUPERHEATER

At least four attributes should be incorporated in the design of a superheater. These are freedom from liability to burn, proper distribution of steam circulation, accessibility for inspection, both internally and externally, and provision for freedom of expansion and contraction.

The various designs of superheater have characteristics distinctly their own. One design that differs materially from all others is the Foster. It is a combination tending to cause leaky joints. As a consequence, the necessity for rerolling a superheater tube is of very rare occurrence.

The upper box or manifold is connected to the steam space of each drum by a steel pipe passing through the bottom of the drum and fixed in position by an expanded joint in the superheater header and by a pad riveted to the drum. Outlet pipes are attached to the lower superheater header and, passing around the steam and water drums are connected over the top of the boiler by a heavy flanged fitting to the main steam outlet.

In addition to the safety valves connecting directly to the boiler drums, a special steel-body safety valve, made to withstand the action of the superheated steam, is connected to the outlet fitting. In order to give access to each expanded joint of the connecting pipes, circular handhole openings, closed by inside caps, are placed in the superheater headers. This superheater is supplied with external flooding pipes connected at one end to the rear head of a steam and water drum below the water line and to the end of the bottom superheater header, Fig. 7. These pipes are arranged so that the superheaters may be drained before cutting the boiler into the steam line. In operation, the steam is taken from the steam space in the boiler drum through dry pipes and enters the superheater through the inlet pipes. The steam on the boiler drums so that when the load is suddenly thrown off the boiler the superheated safety valve opens and causes a flow of steam through the superheater until the fires can be checked,

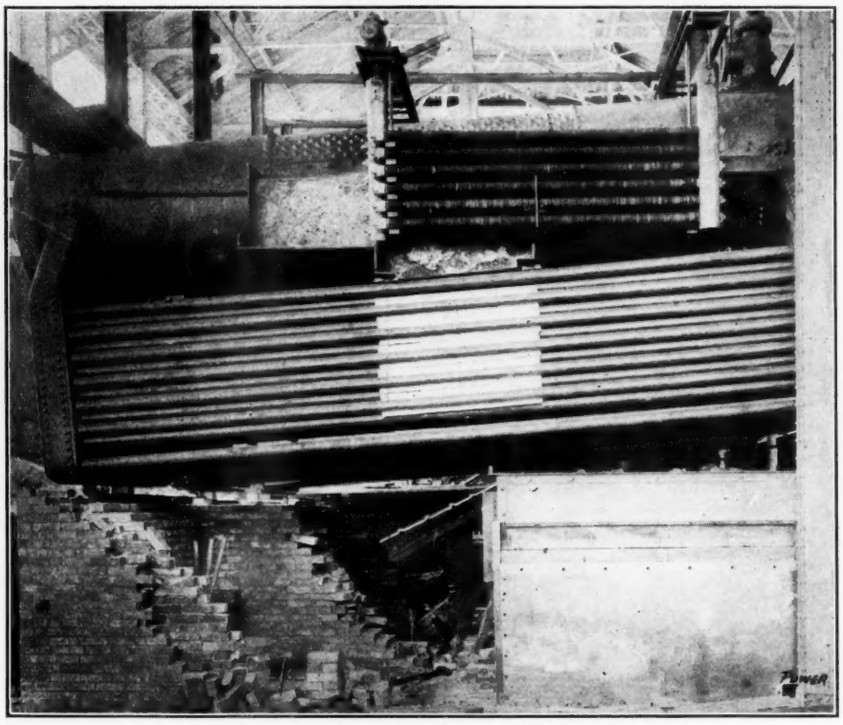


FIG. 16. HEINE BOILER AND FOSTER SUPERHEATER

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tion of annular cast-iron flanges and seamless-steel tubes. The elements or tubes are straight and are generally placed parallel to each other. A manifold joins the elements at one end and the other end of each element is joined to a return header, as shown in Figs. 9 and 10.

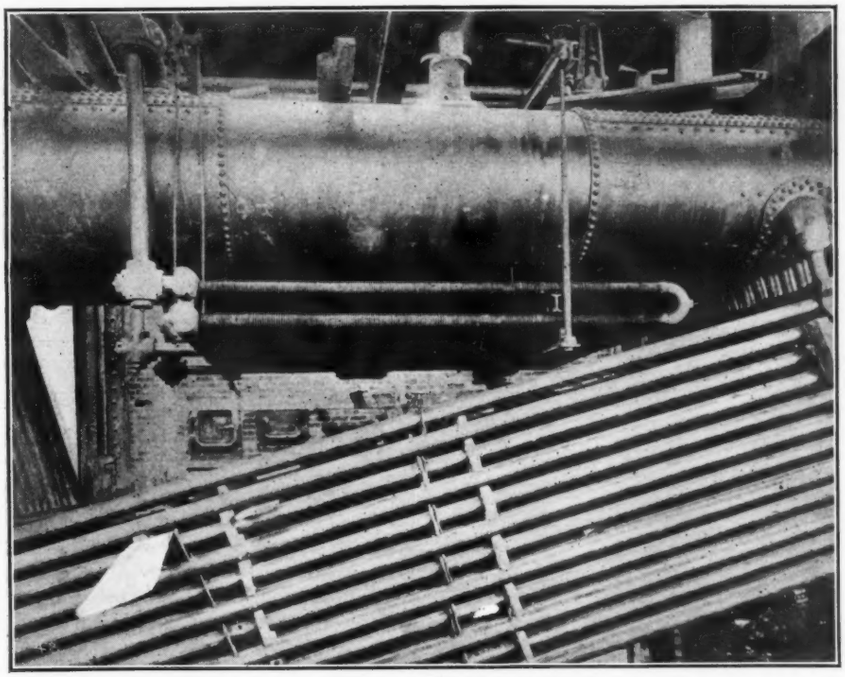


FIG. 17. BABCOCK & WILCOX BOILER AND FOSTER SUPERHEATER

The construction of the combination casting and steel tubing is shown in Fig. 9. The cast-iron flanges fit over the steel tube and are used to protect the tubes from the high temperature of the furnace gases. As the cast-iron rings are shrunk on the tubes, the rings and tubes practically act as a unit. An additional benefit derived from using cast-iron rings is that they act as a reservoir of heat, and are therefore capable of continuously imparting practically the same amount of heat to the superheater, thus maintaining a constant temperature of steam regardless of the ordinary fluctuation in the temperature of the hot gases.

Inside of the steel tubes of each element is a wrought-iron tube. It is kept central in the outer tube by knobs spaced through the length of the inner tube. This feature is shown in Fig. 9. Steam, however, does not enter this inner tube, as it is closed at both ends, as shown in Fig. 11, which is a cross-section through a return header. An exterior view of the same header and handhole plugs is shown in Fig. 12. The purpose of this inner tube is to force the steam as it passes the superheater to go between the inner and outer tubes in a thin sheet. This causes the steam to cling to the heating surface of the outer tube in its passage through each element.

No provision is made for flooding this superheater, and all the protection necessary to guard against burning the elements when getting up steam is found in the covering of cast iron.

This design of superheater is adaptable for use with any type of boiler. In Fig. 13 it is shown attached to a return-tubular boiler. The elements are set on end, and the steam connections are made at the top. The number of elements are determined by the size of boiler and the degree of superheat desired. The superheater is placed in the chamber at the rear end of the boiler, and in order that the greatest amount of heat may be imparted to the superheater, a firebrick arch is constructed in the combustion chamber, as shown. The path of the gases is over the bridgewall, under the arch and through and around the superheater member before entering the tubes of the boiler. This arrangement is not only suitable for new return-tubular boilers, but also for boilers of the same type that have been in service for some years and cannot carry with safety the necessary steam pressure demanded by the engine. As the temperature of superheated steam does not change the pressure, a boiler generating saturated steam can be made to deliver steam to an engine at a high temperature, but low pressure, by using a superheater.

Where this superheater is used in connection with a water-tube boiler, its shape will vary as the design of the boiler dictates. When used with a Stirling boiler the superheater is suspended by means of suitable U-bolts to the I-beams of the boiler support. The members are placed between the first and second banks of tubes in an inverted position. That is, the return bend is placed at the top and the header at the bottom. The general arrangement is shown in Fig. 14. This brings the superheater in the direct path of the hot gases as they pass from the first to the second bank of tubes. The steam connection from the steam drum to the superheater is placed outside of the setting, likewise the pipe running

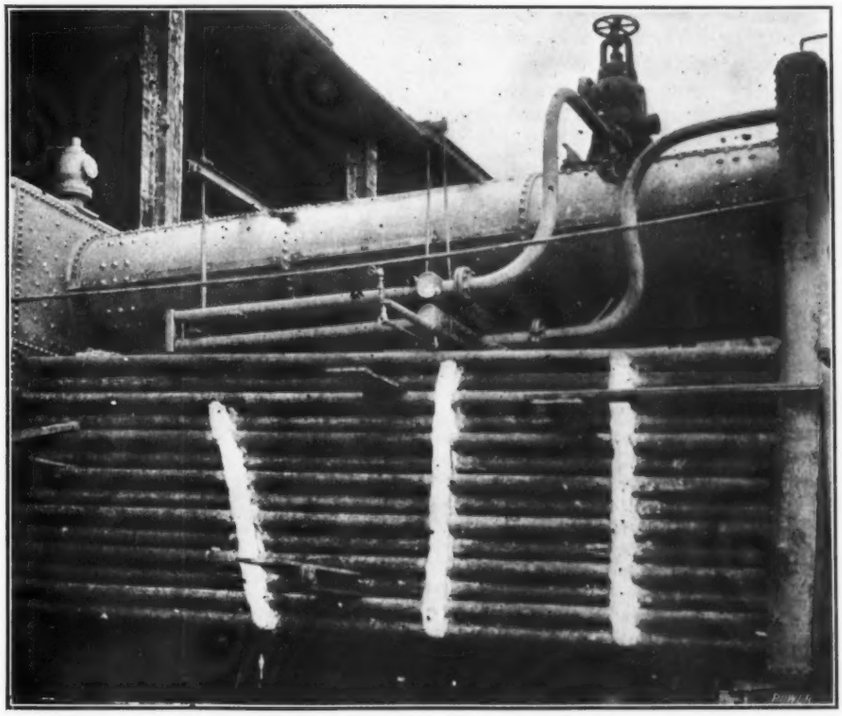


FIG. 18. EDGE MOOR BOILER AND FOSTER SUPERHEATER

from the superheater to the steam main.

This type of superheater is not always made with return bends. What is termed a return header, Figs. 11 and 12, is used with some types of boiler, the design having much to do with the type of header used in connecting the elements.

A Franklin water-tube boiler, Fig. 15, is equipped with this same design of superheater header, as shown in Figs. 11 and 12. Owing to the close proximity of the drum to the tubes it is necessary to place the superheater at the front end and along the side of the drum. The superheater is arranged in a separate chamber, which is divided into two compartments by means of a vertical baffle running transversely across the superheater tubes.

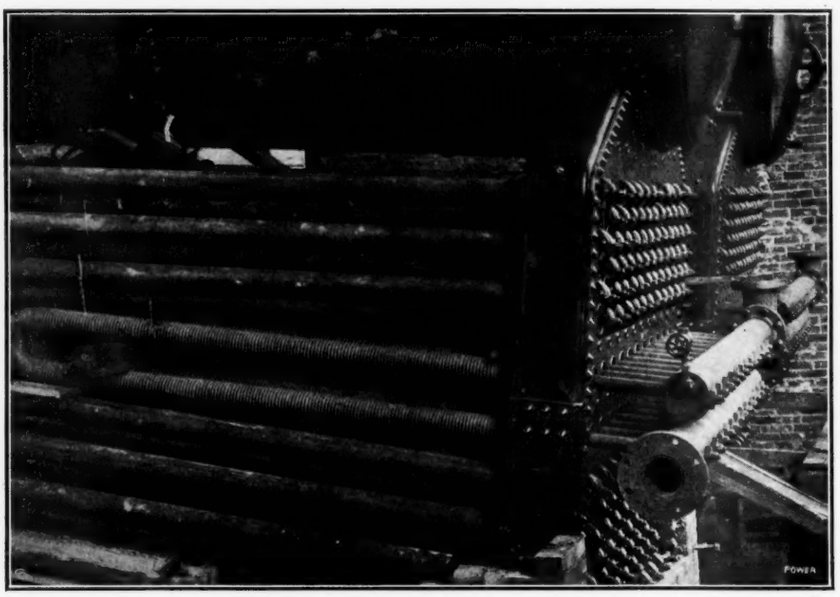


FIG. 19. CASEY-HEDGES BOILER AND FOSTER SUPERHEATER

The bottom of the rear compartment of the superheater chamber is connected by means of a flue in the side walls of the setting directly with the combustion chamber a little to the rear of the bridge-wall. The bottom of the forward compartment of the superheater chamber is in direct communication with the area provided for the passage of the gases passing through the boiler underneath the boiler drums. The circulation of the gases through the superheater chamber is then into the chamber by way of the flue in the side walls, upward among the superheater tubes in the rear compartment, and downward among the superheater tubes in the front compartment of the superheater chamber, from which they issue and join the gases passing through the boiler on their way to the stack. A damper arranged above the vertical baffle in the superheater chamber controls the amount of gases passing through, and consequently the degree of superheat. The amount of hot gas made to pass directly past the superheater is controlled by a damper placed above the top member of the superheater.

Practically the same arrangement of the superheater is made with the Heine boiler, Fig. 16, and other similar designs of boilers, where the superheater is generally suspended from the I-beams by means of suitable bolts.

Fig. 17 shows the method of attaching the superheater to a Babcock & Wilcox boiler. As it is placed central to the longitudinal length of the boiler between the tubes and the drum it lies in the direct path of the hot gases as they leave the first section of tubes and enter the second nest of tubes between the first and steam drums of the boiler is made with second baffle walls. A connection from the the lower header of the superheater on one side of the drum. The steam, after passing through the coils, escapes through a pipe leading from the top header, and on the opposite side of the boiler from the inlet pipe to the main steam header. It is connected to similar types of boilers in practically the same manner. Figs. 18 and 19 show the superheater applied to an Edge Moor and Casey-Hedges boiler, respectively.

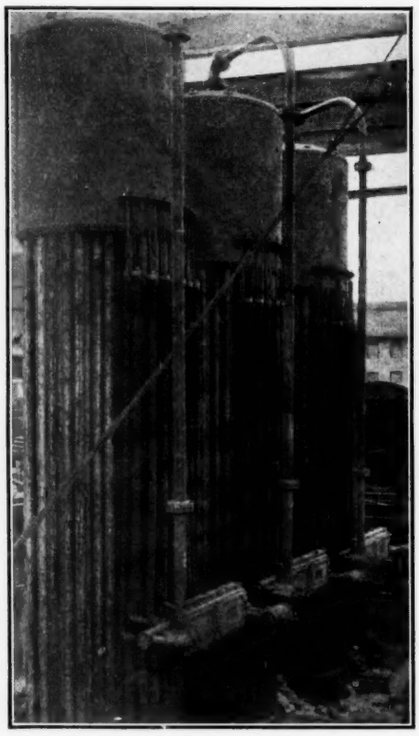


FIG. 20. WICKES VERTICAL BOILER AND FOSTER SUPERHEATER

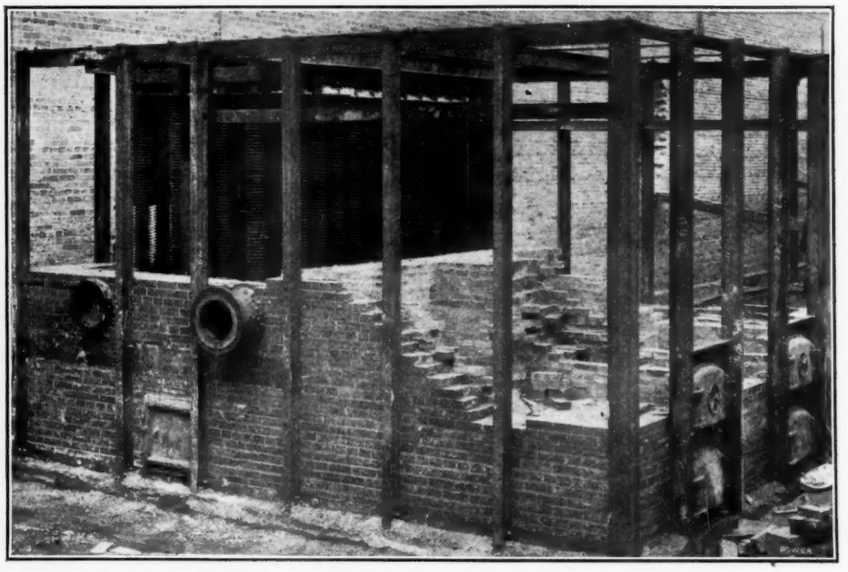


FIG. 21. FOSTER SEPARATELY FIRED SUPERHEATER AND SETTING

This superheater is also adaptable to vertical boilers, as is shown in Fig. 20, which illustrates three Wickes vertical boilers so equipped. The steam pipe, running from the top of the boiler, is shown connected to the superheater header, but the pipe leading from the superheater to

the steam main is not indicated, the connection being shown flanged. The members of the superheater are arranged vertically. The return headers serve as a means of support, which is accomplished by rods being bolted to a cross piece under each return header. The upper end of the bolts pass through holes in an angle piece that is riveted to the boiler.

This superheater is adaptable to separate firing, that is, obtaining the heat for superheating the steam from a furnace separate from the boiler furnace. This design is illustrated in Fig. 21, which shows the method of constructing the framework and the arrangement of the superheating members in relation to the furnace. In steel mills this type of superheater can be operated by waste heat from the various furnaces.

The Steam Turbine in Germany

By F. E. Junge and E. Heinrich

A study of the losses which occur in turbines of the Curtis and Rateau types, with an example from practice on a combination turbine.

Special Discussion of Curtis and Rateau Principles

After having given a general idea in the article in the December 20 number of the methods of investigation we proceed now to the special discussion of the economic qualities of the Curtis and Rateau systems, respectively, by determining, as far as this is possible today by calculating, the losses occurring in each.

Calculation of Losses

(1) By far the largest part of all losses is caused by friction, shock and eddy currents in nozzles and blades. This loss was determined in equation 2 as

$$R = (1 - \eta i)\, h_o \qquad (2)$$

wherein R = loss in heat units per pound of steam, ηi indicated efficiency and h_o available energy of the stage.

(2) *Work absorbed by friction and windage of the rotating wheels.* By the friction between the wheel disks and the surrounding medium a certain amount of work is consumed. Moreover, those blades upon which the live steam does not impinge act as a fan, causing a circular motion of the steam whereby work is again absorbed. Of course, it is not possible to determine the exact amount of this loss by calculation. We profit, therefore, by the results of experimental research, so far as they have been condensed in formulas. For our purpose here we employ the formula of Lasche (Stodola, fourth edition, page 129), which is applicable for admission diameters of from 900 to 1200 millimeters, or 2.95 to 3.94 feet, with length of blades ranging from 10 to 50 millimeters, or 0.394 to 1.97 inches. In this formula the friction and windage expressed in kilowatts for a wheel not inclosed in a casing, and working without admission of steam but in the medium steam, is

$$N_{kw} = \beta \cdot 10^{-9}\, D_m\, n^3\, L_m\, \gamma \qquad (5)$$

wherein D_m is admission diameter; L_m, mean length of blades; n, number of revolutions per minute; γ, specific weight of steam.

If metric units are employed (D and L in meters, γ in kilograms per cubic meter) the coefficient β becomes

17.5 for wheels with one row of buckets,
20.5 for wheels with two rows of buckets,
28.0 for wheels with three rows of buckets,
42.5 for wheels with four rows of buckets.

If British units are employed (D and L in feet, γ in pounds per cubic foot) the coefficient β becomes

26.1 for wheels with one row of buckets,
30.5 for wheels with two rows of buckets,
41.7 for wheels with three rows of buckets,
63.2 for wheels with four rows of buckets.

Recalculated in B.t.u. per second and considering that 1 kilowatt = 737.3 foot-pounds per second and 1 B.t.u. = 778 foot-pounds, this equation reads:

$$V = \frac{737.3}{778}\, \beta \cdot 10^{-9}\, D_m\, n^3\, L_m\, \gamma = 0.948\, \beta \cdot 10^{-9}\, D_m\, n^3\, L_m\, \gamma \qquad (6)$$

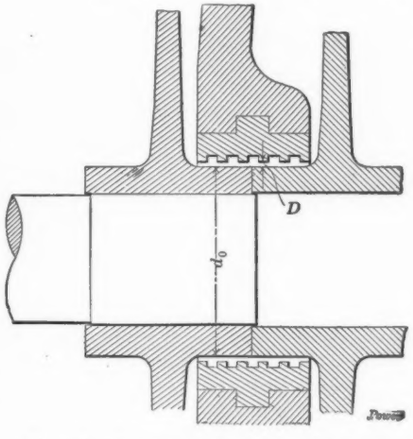


Fig. 11. Stuffing Box of a Rateau Stage

(3) *Losses through leakage on the wheel hubs.* Between the hubs of the wheel disks and the hubs of the rotating wheels there is, of course, a certain clearance through which steam can escape from one stage into the next, without performing any useful work; see Fig. 11. If no resistance were opposed to the flow of steam through these clearances the actual passage velocity of the steam would be equal to the theoretical velocity c_o, which corresponds to the theoretical drop of heat. In order to get small losses one must take care, therefore, that this passage velocity remains as low as is practically possible, or, in other words, the resistance offered to the flow must be as high as possible. This is attained by a labyrinth packing, as shown in Fig. 11. This construction affords a reduction of the actual passage velocity down to 0.6 c_o. If $f = \pi d_o s$ is the cross-section of the orifice of loss and γ the specific weight of the steam issuing, the lost steam weight is:

$$g_o = 0.6\, \pi\, d_o\, s\, c_o\, \gamma \qquad (7)$$

If further h_o equals the available energy in the stage in heat units per kilogram of steam, the loss through leakage may be expressed in heat units as follows:

$$n = g_o\, h_o \qquad (8)$$

In the first stage of an impulse turbine there is no leakage loss on account of the fact that there is no clearance through which steam could escape.

Energy to Be Regained

Part of the velocity of issue c_2 (Fig. 7, December 20), may be regained by catching the steam jet, emanating from the rotating wheel, in the following guide blades in such a manner that as little as possible of the inherent velocity of the steam is destroyed. It is customary to assume that with average quality of design and shop work, about 60 per cent. of the residual energy $\frac{c_2^2}{2g}$, of the steam is utilized in the following guide blades. Thus the energy regained per kilogram of steam passing through the blades is in heat units:

$$M = 0.6\, \frac{c_2^2}{2\, g\, 778} \qquad (9)$$

Combination of Losses

After having determined the various losses and the energy to be regained we combine the results in the following formulas: If g pounds is the weight of steam flowing through the turbine per second, we must bear in mind that not the whole of it passes through the guides and moving blades, but that the amount g_o, passing through the clearance on the hubs, must be subtracted, so that $g - g_o$ pounds per second pass through the blades rendering useful work. It must be further considered that the loss R

and the amount M of the regainable energy refer to the unit of steam weight which flows through the guiding and moving blades, while the calculated amount for windage V and for leakage losses U refers to the total steam weight flowing through the turbine. If we designate by L the sum of losses per pound of steam,

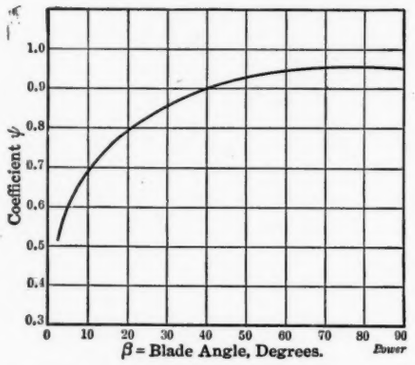


FIG. 12. COEFFICIENT OF RESISTANCE OF ROTATING BLADES

the total loss in the turbine is: $L \times g$, and we get the following equations:

$$L\,g = (g - g_o)\,R + U + V - (g - g_o)\,M$$
$$= (g - g_o)(1 - \eta_i)\,h_o + g_o h_o + V - (g - g_o)\,M = g\,(1 - \eta_i)\,h_o + g_o h_o \eta_i + V - (g - g_o)\,M \qquad (10)$$

Hence the loss per pound of steam is found to be:

$$L = (1 - \eta_i)\,h_o + \frac{g_o}{g}\eta_i h_o + \frac{V}{g} - M \qquad (11)$$

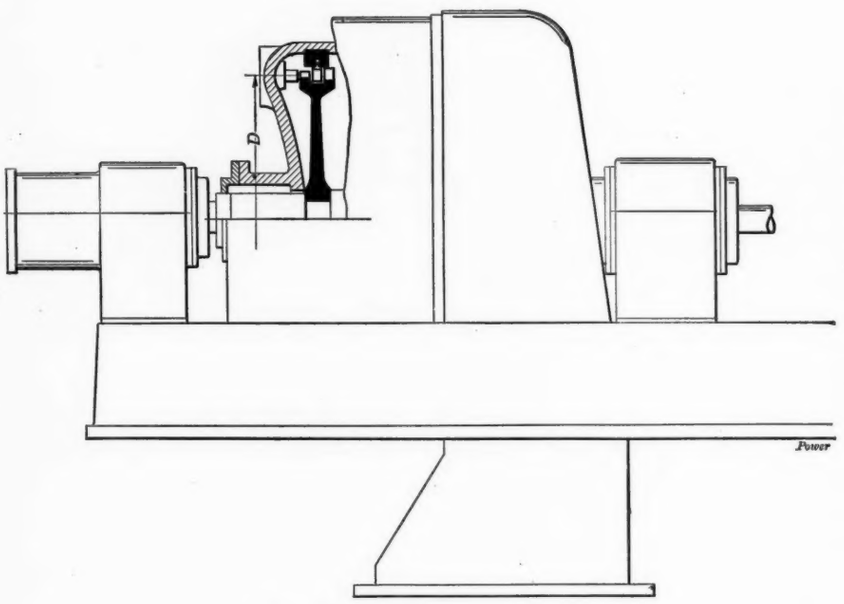


FIG. 13. SINGLE-STAGE CURTIS

In the above expression we have assumed the factor of M, $\frac{g - g_o}{g}$, to be equals 1 for the sake of simplicity, because the total amount of M is comparatively small and because g_o is a small amount compared to g. For the first stage of a turbine, where leakage losses do not occur, we get:

$$L_1 = (1 - \eta_i)\,h_o + \frac{V}{g} - M \qquad (12)$$

With Curtis wheels the residual exit velocity is, as a rule, not utilized for reasons of design and because it is very small. Hence, we have for Curtis wheels:

$$L_c = (1 - \eta_i)\,h_o + \frac{V}{g} \qquad (13)$$

THE COEFFICIENTS OF LOSSES

Before attempting to carry through a calculation of an example on the basis of these reflections it appears necessary to determine the amount of the coefficients of loss, ϕ, ψ, ξ, wherefore we profit again by the latest experimental researches, especially of Stodola. As to coefficients it is to be noted that the value depends essentially on the state of the surface of the nozzles or blades, which is to say, it depends preëminently on shop work. Further, it is to be noted that the nozzles and blades may vary the state of their surface even during operation, as a result of corrosion of the blades and of sediment of impurities of the steam. Hence, the coefficients of resistance of

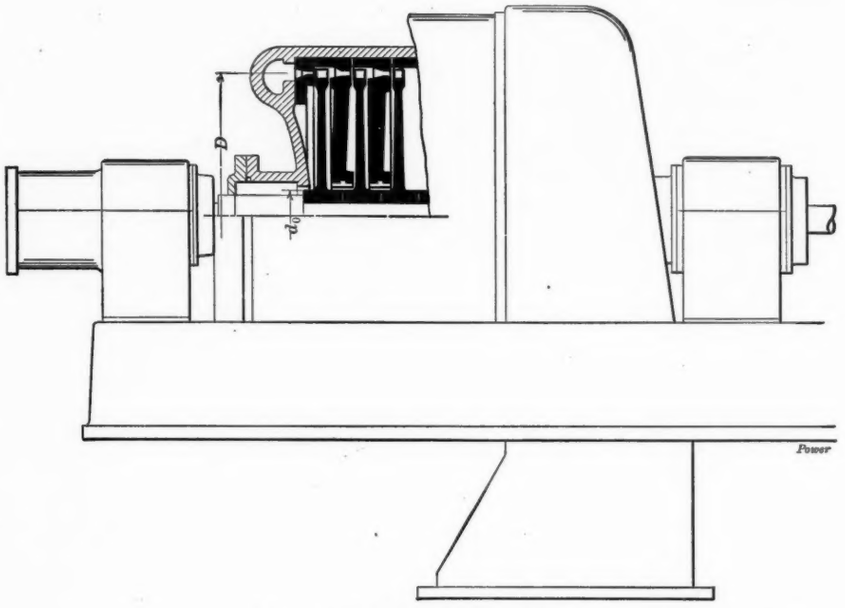


FIG. 14. THREE RATEAU STAGES

one and the same turbine are far from constant.

The coefficient ϕ is usually assumed to be 0.95 for first-class shop work. As to the coefficient ψ the results of the experiments of various investigators differ essentially. Thus, for instance, Briling and Rateau found that the coefficient ψ increases with the relative velocity w_1, hence the loss of energy decreases, while Stodola and Huguenin found, in contradistinction, that ψ decreases with w_1. On the other hand, all experimenters have established the fact that ψ decreases with increasing curve of blade, that is to say, with decreasing blade angles. With this consideration in mind we assume according to Stodola that the coefficient of resistance is independent of the steam velocity w_1 and depends only upon the curve or angle of the blades, as shown in Fig. 12.

The number ξ in equation 4, representing the loss in the first row of blades, in percentage of the energy of issue $\frac{w_2^2}{2g}$, is assumed to be 25. This value was determined from an analysis of a Curtis turbine made by Stodola.

EXAMPLE FROM PRACTICE

In a turbine of 1000 kilowatts at 3000 revolutions per minute, consisting of eight

simple-pressure stages (Zoelly system), the high-pressure part is to be replaced by one Curtis wheel. Figs. 13 and 14 show the two devices. In order to utilize by means of the original Rateau principle the same drop of energy as in one Curtis wheel we must assume at least three Rateau stages as its equivalent. Furthermore, the mean wheel diameter and therefore the circumferential velocity shall remain the same. In our example which is taken from actual practice the admission diameter is 1 meter = 3.28 feet. The operating conditions of the turbine are:

Admission pressure = 180 lb. per sq.in. absolute.
Steam temperature = 525 deg. F.
Superheating = 157 deg. F.
Vacuum = 94 per cent.
Steam consumption = 15 lb. per kilowatt-hour.
Steam weight per sec. = 4.17 lb.

In the high-pressure part the steam expands down to 33.5 pounds per square inch. From the Mollier diagram is obtained a theoretical heat drop of 146 B.t.u. The clearly defined conditions of the comparison of the two systems are, therefore, one Curtis wheel as against three Rateau stages; equal circumferential velocity in both cases; equal heat drop to be utilized in both cases; coefficients of losses determined by same laws.

Curtis Wheel

Fig. 15 shows the Mollier diagram, Fig. 16 the velocity diagram. In the latter such angles have been used as obtain in practice.

The theoretical exit velocity is $c_0 = 223.8\sqrt{146} = 2705$ feet per second.

The circumferential velocity is $u = \frac{3.28\,\pi \cdot 3000}{60} = 515$ *feet.*

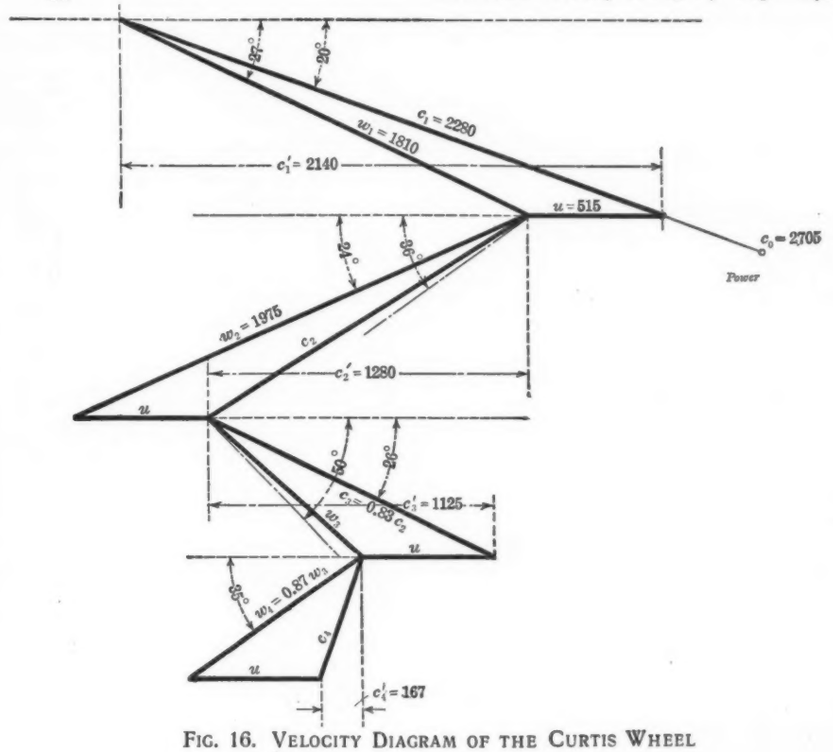


Fig. 16. Velocity Diagram of the Curtis Wheel

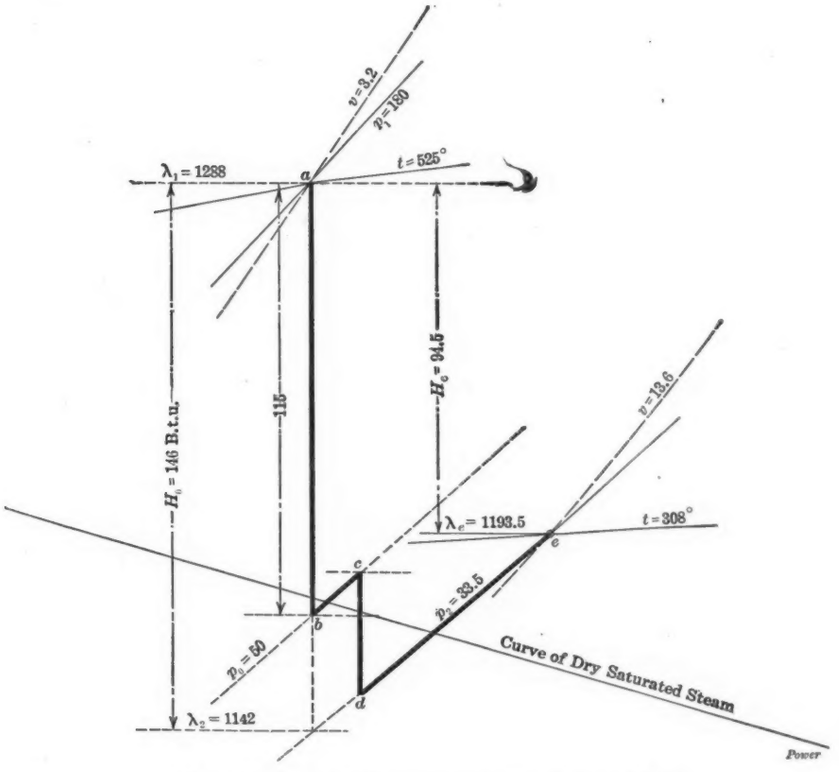


Fig. 15. Mollier Diagram of the Curtis Wheel

In accordance with the above we do not expand in the nozzle down to the counterpressure 33.5 pounds per square inch, but to the pressure 50 pounds per square inch. Then we find from the Mollier diagram that in the nozzle 115 B.t.u. are converted, corresponding to a theoretical velocity of $223.8\sqrt{115} = 2400$ feet per second and an actual velocity of $0.95 \times 2400 = 2280$ feet per second, as shown in the diagram. In the nozzle we expand theoretically down to point *b*, but owing to the losses occurring the end point of the expansion is shifted to *a*.

[Loss 115 $(1 - 0.95^2) = 11.5$ B.t.u. per pound.]

In the first row of revolving blades we expand further down to *d*, realizing a drop of heat of $h_2 = 32$ B.t.u. From the velocity diagram is found $w_1 = 1810$ feet per second, and w_2 from equation 4 in the December 20 number.

$$w_2 = \sqrt{\frac{w_1^2 + 2\,g\,778\,h_2}{1 + \xi \div 100}} = \sqrt{\frac{1810^2 + 2 \times 32.185 \times 32}{1 + 0.25}} = 1975$$

feet per second

Having w_2 we get from the velocity diagram c_2. This is the velocity with which the steam enters the deviating blades. For the latter we obtain for the smallest blade angle (26 degrees) the coefficient from Fig. 12, $\psi = 0.83$. And the velocity of the steam leaving the deviating blades, being equal to the entrance velocity of the steam into the second row of revolving blades, becomes $c_2 = 0.83\,c_2$.

The velocity conditions in the second row are normal and need not be further explained. It may be mentioned that to the smallest angle of the second row of

revolving blades, 35 degrees, corresponds a coefficient $\psi = 0.87$. The respective projections of the absolute entrance and exit velocities upon the direction of travel are obtained directly from the velocity diagram.

The indicated efficiency, according to equation 4, is

$$\eta i = \frac{2 \times 515 \times (2140 + 1280 + 1125 + 167)}{2705^2} = 0.663$$

Therefore the friction loss per pound of steam is

$$R = (1 - \eta i)\, ho = (1 - 0.663)\, 146 = 49.2\ B.t.u.$$

It remains to determine the windage from equation 6. The mean length of blades has been assumed as 5/8 inch = 0.052 feet. The mean diameter d = 3.28 feet, the specific weight from the Mollier diagram $\gamma = \frac{1}{13.6} = 0.0735$ pound per cubic foot, the coefficient $\beta = 30.5$. Thus we get the following formula:

$$V = 0.948 \times 30.5 \times 10^{-9} \times 3.28 \times 3000^2 \times 0.052 \times 0.0735 = 9.8\ B.t.u.$$

and per pound of steam per second $\frac{V}{g} = \frac{9.8}{4.17} = 2.3$ pounds of steam per second.

Hence the total loss according to equation 13:

$$L_c = 49.2 + 2.3 = 51.5\ B.t.u.\ per\ lb.\ of\ steam$$

and the interior efficiency of the Curtis wheel is:

$$\eta = \frac{146 - 51.5}{146} = 0.647$$

Uncle Pegleg's Philosophy

The old man explains the difference between force and work, shows how the resultant of two forces may be obtained, and incidentally works out some problems in proportion.

"I started in the other day to explain something to you and you led me off," said Uncle Pegleg, another day.

"So?" I said. "What was it?"

"When I asked you about the pull of

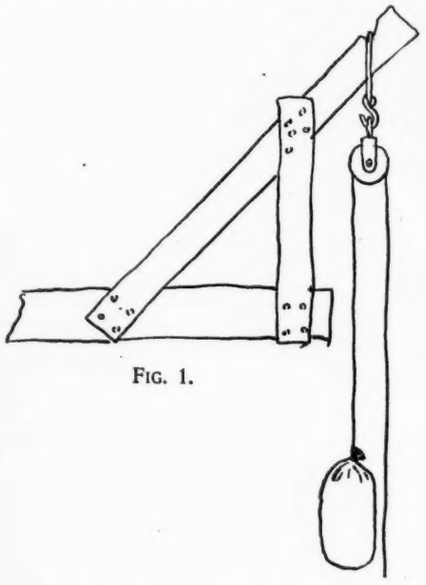


FIG. 1.

that bag of gravel they were hoisting, we got off onto the pull on the rope. I wanted to get at the stress on the strut."

"What do you mean—the stress on the strut?"

"They had a board stuck up like this (Fig. 1) with a pulley on the end of it to hoist the gravel with. Well, I want to know what is the stress on that strut. How hard does it push on the nails that hold it? Take this case," and he drew Fig. 2. "Suppose the weight is 100 pounds. How would you go to work to find the force with which the boom was pushing down into the corner *A*?"

"Two hundred pounds, isn't it?" I said on a guess.

"No, because the pull in the part of the rope between *B* and *C* is 100 pounds, but it isn't pulling in the direction of the strut; and the pull on the piece of rope between *C* and *D* is 100 pounds, but that isn't pulling in the direction of the strut either. A pull that don't pull in the right direction may help some, but not its full amount."

"Well, what's the answer?"

"Suppose a boat was going across a river and the man in it rowed straight for the opposite bank all the time. If he went with a steady, uniform speed of 200 feet a minute, he would be here (indicating *a* in Fig. 3) at the end of the first

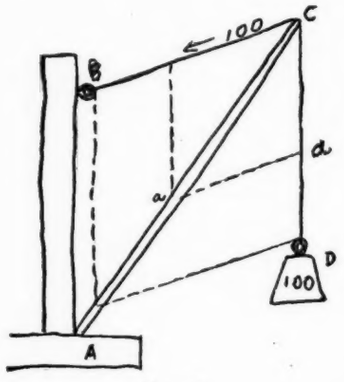


FIG. 2.

minute, here (indicating *b*) at the end of the second minute, here (*c*) at the end of the third minute, etc."

"If the current didn't carry him down," I said.

"Exactly. That's just what I was coming at. If the current carried him downward at the rate of 100 feet a minute, he would be at *d* instead of at *a* at the end of the first minute, just as though he had gone straight to *a* by reason of his rowing and then to *d* by reason of the current. At the end of the second minute he would be at *e* instead of at *b* and at the end of the third minute at *f* instead of at *c;* and always supposing that the velocities were uniform, the path that would have actually followed would be *o d e f*. Is that plain?"

I admitted that it was.

"Well, then, if *o a* is proportional to his velocity in the direction *o a*, i.e., across the river, and *a d* is proportional to his velocity in the direction *a d*, i.e., down the river, *o d* must be proportional to his actual velocity, because he actually goes from *o* to *d* in the same time that the other velocities would have taken him from *o* to *a* or from *a* to *d*."

"What has that got to do with the force on the strut?" I asked.

"Everything. We will come to that. I am showing you now how, if you have two velocities and their direction, you can find the actual velocity and direction which they, acting together, will produce. The same thing applies to forces. Here (Fig. 2) you have two forces; one acting in the vertical direction *C D* and the other acting in the direction *B C*. You want to find what their resultant in the direction *C A* is. You do it the same way as with velocities. Now let us see what we do.

"Starting from the starting point *o*, lay off a line *o a*, Fig. 4, two inches long for the 200-foot velocity across the river, and from the same point a line *o g*, one inch long, for the velocity down the river. Com-

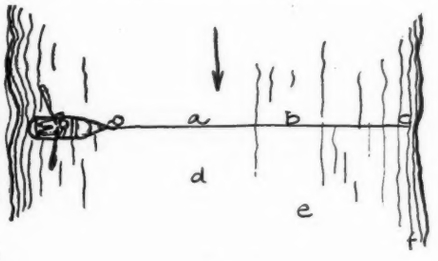


FIG. 3

plete the parallelogram of which these are two sides by drawing in the sides *g f* and *a f;* then the diagonal *o f, drawn from the starting point o,* will be the actual or resultant direction and velocity."

"Wouldn't it be just the same if you took the diagonal *a g*?" I asked.

"No, because the corners may not be right angles. It only happened so in this case because the man was rowing at right angles to the current. Let's see what would happen if he started up the river at an angle of 30 degrees."

With the 30-degree angle of his drawing set he drew Fig. 5. The line *o a* points 30 degrees up stream. The line *o g*, one-half as long, because the velocity is one-half as great, points down stream, representing the direction in which the boat is carried by the current. Complete the parallelogram. Then the diagonal *o f* from the starting point is proportional in length and represents by its direction the actual velocity and direction which the boat would take. You can see that the other diagonal *a g* would be away off. Always start at the starting point to draw your diagonal. You could have done it just as well by drawing the line *o a* and then *a f* and connecting *o* to *f*, using only the triangle *o a f* instead of the parallelogram. They call this the triangle of forces or velocities, but if you ever get confused, go back to the starting point, put in both velocities or forces from that point, make your parallelogram and use the diagonal from the starting point and you will be all right.

"Now, then, let's see about the force. Here (Fig. 2) you have equal forces acting in the directions *C B* and *C D*. Lay off equal distances, since the forces are equal on these lines and complete the parallelogram drawing in the dotted lines. Then the force acting in the direction *A C* will be as much greater than 100 pounds as *C a* is longer than *C b*. If you make *C b* 1 inch long to equal 100 pounds and *C a* is 1.25 inches, then the force acting in the direction *C a* will be 125 pounds. If the force was any other number, 140 for instance, you would have to do it by proportion. Know how to do proportion?"

'I didn't.'

"Well, it's easy. The old rule of three. Come up to the house and I will give you an arithmetic. You can learn the whole section on proportions in an evening and they are always coming up. You know

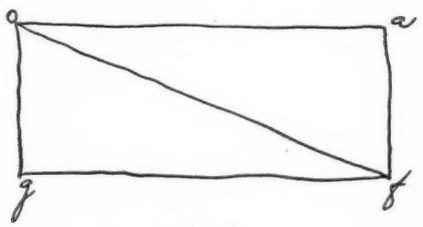


FIG. 4.

three things and you want to know a fourth. For instance, in this case we know the length of the line *C b* and of the line *C a* and we know the force acting in the direction *C D*, say 140 pounds. We know that this force bears the same relation to the force acting in the direction *C A* that the length of the line *C d* does to the length of the line *C a*.

Set down the two similar terms that

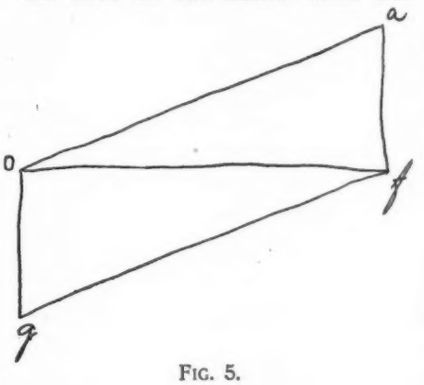


FIG. 5.

you know; in this case the two lengths of the lines, 1 inch and 1.25 inches, putting the one that agrees with the odd term that you know first

1 . 125 :: 140 : x

This reads, as one is to 1.25 so is 140 to the quantity you want to know.

"The thing to look out for is to get them so that the two quantities to the left of the double colon will bear the same relation to each other as those to the right and in the same order. Thus,

2 · 4 :: 3 : 6

You know that 2 is one-half of 4 and 3 is one-half of 6. If you get them in this order it will be true that the product of the two inside figures will be the same as that of the two outside. Try it."

I saw that $2 \times 6 = 12$ was the same as 4×3.

"If you have any three of them you can find the other," continued my instructor. "If one of the end ones is missing, multiply the two middle ones together and divide by the end one which you have. If one of the middle ones is missing, multiply the end ones together and divide by the inside one which you have. Simple enough, isn't it? In the case of the strut we have the two middle terms given, 1.25 and 1.40, and one end term, 1.

$$\frac{1.25 \times 140}{1} = 175 \text{ lbs.}$$

"That proportion rule is one of the handiest things in the trade," continued the old man. "All you've got to do is to look out and get them set down right.

"If you put the smaller of the pair of known terms first, fix it so that the smaller of the pair of unknown terms will come inside, i.e., will come first on that side. For instance, you know that the horsepower of an engine is proportional to the piston speed. If it runs twice as fast, other things being equal, it will develop twice as much horsepower. Well, suppose an engine develops 330 horsepower at 120 revolutions, how much will it develop at 125?

Here your pair of similar terms are the revolutions 120 and 125; if you put them down in that order, you must put the 360 first in the next pair because the 360 goes with the 120 of the first pair.

120 : 125 :: 360 : hp

$$\frac{125 \times 360}{120} = 375 \text{ hp}$$

"But sometimes it happens that a proportion is 'inverse' or backward. You know that the smaller pulley you put onto a driven shaft the faster it will run. The speeds of the shafts are inversely proportional to the diameters of the pulleys. A shaft running at 180 revolutions per minute carries a pulley 36 inches in diameter which is belted to a pulley 24 inches in diameter on another shaft. How fast will the other shaft run? Here your known pair are the diameters 24 and 36; the known term of the unknown pair is the 180 revolutions. Well, you put it down so

24 : 36 :: 180 : revs.

or so

24 : 36 :: revs. : 180

You know that the revolutions will be greater than 180 with the 24-inch pulley, as much greater than 180 as 36 is greater than 24, so that you can see that the first way is right and that whereas with a direct proportion you put the known term of the incomplete pair first if its corresponding term was first in the other pair (120 revolutions for 360 horsepower both first in the other example). You now put them just the other way.

revs revs hp hp
120 : 125 :: 360 375

diam. diam revs revs.
24 : 36 :: 180 270

$$\frac{36 \times 180}{24} = 270 \text{ revs.}$$

"How much do you suppose that crosshead bears on the guide," asked the old man, coming back to the main question, after this little lesson in arithmetic, "when the crank is at an angle of 45 degrees?"

It was too deep for me even with the explanation which I had heard, so I let him figure it.

"Let's lay it out," said he. "Give me that little drawing board and the tools."

I tacked a clean sheet of paper on his small board and he laid the T-square on it and drew the center line *A B*, Fig.

6. On this he drew a circle for the path of the crank pin, put in the crank $O\,C$ at an angle of 45 degrees with the line of centers, and drew the connecting rod $C\,D$, twice the diameter of the crank-pin

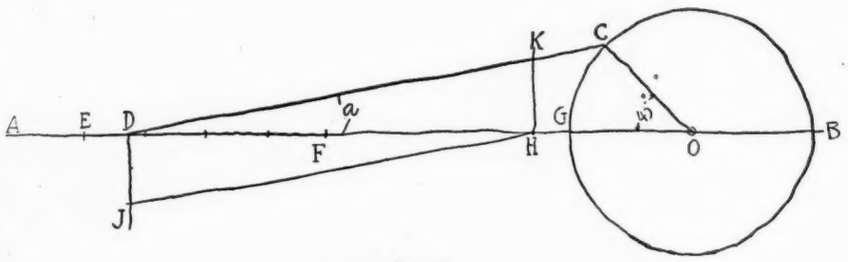


FIG. 6.

circle, for in our engine the connecting rod was twice the length of the stroke. Setting off the same distance from G, he determined the point E where the wristpin would be when the crank was at G, and from B the point F where the wristpin would be when the crank was on the farther center. He then showed me that the crosshead would travel from E to F and that it would be at D when the crank was at 45 degrees.

"There's another example in proportion for you," he said. "I've made the stroke here $E\,F = 3$ inches. The piston has traveled $E\,D =$ about 9/16 inch. Our indicator takes a diagram 3¾ inches long. As the distance $E\,D$ is to the distance $E\,F$ so is the distance of the point on the diagram which represents the position of the piston at that point of the stroke to the length of the diagram.

$ED : EF :: X :$ length of diagram

$\frac{9}{16} : 3 :: X : 3\frac{3}{4}$

Multiply the two outside terms together.

$3\frac{3}{4} = \frac{15}{4}$

$\frac{15}{4} \times \frac{9}{16} = \frac{135}{64}$

Now divide this by the inside term.

$\frac{135}{64} \div 3 = \frac{45}{64}$

"Good. Now measure out on one of your diagrams 45/64 of an inch and see how much pressure you have between the forward and back-pressure lines at that point."

I measured one of the diagrams and found about 72 pounds.

"That," went on the old man, "is the difference between the pressures on both sides of the piston when the crank is at C—on each square inch of it. How many square inches are there?"

It was an 18-inch cylinder and I found in the table of areas that it had 254 square inches. Multiplying this by 72 I found that the unbalanced push on the piston was

$72 \times 254 = 18{,}288$ lbs.

"Yes, over nine tons," said the old man. "Pretty good shove, eh? Now, this push acts on the wristpin D in the direction $D\,B$. Let's lay off $D\,H$, say 5 inches, to represent it. This force is split up into two forces, one that acts through the connecting rod in the direction $D\,C$, and one that pushes the crosshead down onto the guide in the direction $D\,J$. Now, if we draw $H\,J$ and $H\,K$ parallel to $D\,K$ and $D\,J$, we shall have a parallelogram of forces of which $D\,H = 18{,}288$ pounds is the diagonal, and the downward force on the guides will be the same part of 18,288 pounds that $D\,J$ is of $D\,H$, and the shove on the rod will be as much greater than 18,288 pounds as $D\,K$ is greater than $D\,H$. You can scale it off. For instance, $D\,H$ is 5 inches and represents 18,288 pounds. $D\,J$ is about ⅞ of an inch.

$DH : DJ :: 18{,}288 : X$

$5 : \frac{7}{8} :: 18{,}288 : X$

$\frac{\frac{7}{8} \times 18{,}288}{5} = \frac{\frac{7}{8} \times \overset{2286}{\cancel{18{,}288}}}{5} = 3200$ lbs

or over a ton and a half.

"If you want to get it more accurate than you can draw and scale it, you can calculate it."

"How?" I asked.

"You have a triangle $O\,C\,D$ of which you know the length of two sides and one of the angles. A triangle has three sides and three angles. If you know any three of these six properties you can find the rest, but one of the known properties must be a side if you want to get actual lengths. You can get the proportions of the sides if you know only the three angles but not the actual lengths, for a triangle of the same shape may be so small you would need a microscope to see it or as big as all outdoors. Opposite each side of a triangle is an angle. The angle opposite the side $C\,O$ is the angle at D which we don't know. The angle opposite the side $C\,D$ is that at O, 45 degrees.

"The sides of triangles are proportionate to the sines of their opposite angles. Then, calling the unknown angle at the sharp point of the triangle a,

$CD : \sin 45° :: CO : \sin a$

Look up a table of sines in that handbook."

I passed him the book open to the table of sines and he showed me that the sine of 45 degrees is 0.70711.

"Now we know that the connecting rod $C\,D$ is 4 times the length of the crank $C\,O$. Call $C\,D$ 4 and $C\,O$ 1; then

$4 : 0.70711 :: 1 : \sin a$

Multiply the two middle terms together and divide by the known end one.

$\frac{0.70711 \times 1}{4} = 0.17678$

This is the sine of the angle a. Hunt it up in the column of sines."

The nearest that I could find to it was 0.17794.

"That's all right. This table goes by quarter degrees or 15 minutes. That's near enough for our purpose. If we were working astronomy we should have to use finer tables. The sine value 0.17794 corresponds to 10 degrees and 15 minutes. See?"

That was as easy as looking up areas or circumferences.

"Now," continued the old man, "while your've got that angle there see what its tangent is."

I looked in the tangent column on the same line and found 0.18083.

"You don't know what a tangent is, do you?" he asked.

I had heard of things "going off at a tangent" and had a shady idea that it was a straight line hitched onto a circle.

The old man drew Fig. 7. "Here is a piece of a circle," he said, "drawn with a certain radius $O\,A$. Draw a line as $O\,B$ from the same center and it will include a certain angle. Draw a line perpendicular to the end of the radius up to the line $O\,B$, bounding the angle, and it will

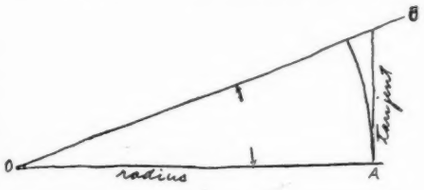


FIG. 7.

be the tangent of that angle. The table tells you what the length of the tangent would be if the length of the radius were unity or 1.

"Well, $H\,K$ (Fig. 6) is the tangent of the angle a with a radius of $D\,H$; that is to say, for an angle of 10 degrees 15 minutes $K\,H$ is 0.18083 of $D\,H$. Then, since $D\,H$ represents 18,288 pounds, $K\,H$ represents

$18{,}288 \times 0.18083 = 3292$ lbs.

The Influence of the Cylinder Wall

By Prof. V. Dwelshauvers-Dery

In which it is demonstrated that the thermal influence of the cylinder walls depends very little upon the size of the machine and very much on the conditions of operation.

The theory expounded by Professor Heck in the issue of September 13 last of POWER, under the title "Some Points Favoring Compression," is a complete abstraction of the disturbances brought about in the evolution of the steam by the thermal action of the metallic walls, which, during all the cycle, exchange heat with the steam. It seems that according to his idea this disturbing action will not be of importance except in small machines such as that with which I have operated at the laboratory of the University of Liège—diameter 12 inches, stroke 24 inches, 30 horsepower—but will be negligible for the larger machines, such as are met with in industrial use.

I do not know on what duly established facts he rests this hypothesis. I would like to believe that it is upon experiments made with the same precision as those made at the laboratory of Liège, and I should like to be assured upon this point. Meanwhile, I will try to demonstrate, contrary to his assertion, that the extent of the thermal influence of the cylinder walls depends very little upon the size of the machine and, on the contrary, very much on the conditions of operation. It is only the efficacy of the steam jackets which is reduced in large cylinders, but the evil or beneficial effects of the degree of admission of superheat and of high speed are as marked in the large as in the small machines.

It is this that I shall show, depending upon tests made with the greatest care, by different experimenters, and recorded in various periodicals, notably:

A. Bulletin de la Société Industrielle de Mulhouse Alsace. 1. 1876. Report of Hallauer on eight experiments made in 1873 and 1875 on the famous engine of Logelbach under the direction of G. A. Hirn, by Hallauer, Grosseteste and V. Dwelshauvers-Dery. 2. 1888. V. Dwelshauvers-Dery. New method for representing the exchanges of heat between the metal and steam. 3. 1889. V. Dwelshauvers-Dery. Exposition of the experimental theory of Hirn for single-cylinder machines.

B. Act of the International Congress of Applied Mechanics, held at Paris, September 16 to 21, 1889. 4. V. Dwelshauvers-Dery. Note on various methods of economizing steam in single-cylinder machines.

C. Excerpt from the minutes of the proceedings of the Institution of Civil Engineers, London, Volume XCVIII, Session 1888-1889. 5. Bryan Donkin and V. Dwelshauvers-Dery. Heat expenditure in steam engines.

Of the experiments with which these memoirs deal, seven have been made by Bryan Donkin on a small experimental machine of six indicated horsepower; diameter, 8.4 inches; stroke, 14 inches; revolutions per minute, 90; real degree of expansion, 0.684; clearance, 0.10 of the piston displacement. This engine was provided with a gas-flame jacket, and the object of the test was to compare the performance of the engine with and without the jacket. Of these seven tests we will retain in that which follows only the two carrying the numbers two and four respectively, made without jackets, the first condensing and the second noncondensing; the first on August 9, 1888; the second, July 26 of the same year.

Twelve have been made by Willans, the celebrated English engine builder, in order to determine the effects of differences in speed. They are classed in four series of three tests each: the first with the point of cutoff at 0.437, the second at 0.339, the third at 0.264 and the fourth at 0.216; and in each series the first test at about 400 turns per minute, the second at 200, the third at 100, all noncondensing. The engine is single acting; diameter, 13 inches; stroke, 6 inches; indicated horsepower varying between 35 and 6; clearance, 0.07 of the piston displacement. We will consider here only the three tests of the fourth series, Nos. X, XI and XII, for which the real degree of expansion was 0.28.

Eight tests were made by Hirn on the celebrated engine of Logelbach, operating ordinarily with superheated steam: two in 1873, one November 18 with superheated steam, the other November 28 with saturated steam; six in 1875, all with superheated steam and all condensing, save one, that of October 28, which was noncondensing, with degrees of expansion varying between 0.162 and 0.457. Of these eight tests we will retain only three under the respective numbers II, V and VI: No. II, of November 28, 1873, condensing and with saturated steam, with an actual degree of expansion of 0.257; No. V, of September 7, 1875, condensing, superheated steam, real degree of expansion 0.162; No. VI, September 8, 1873, condensing, saturated steam, real degree of expansion 0.162.

The dimensions are: Diameter, 24 inches; stroke, 70 inches; clearance, 0.01. The number of revolutions did not vary widely from 30 per minute, and the output was 150 horsepower. The object of the test was multiple. It bore principally on the effect of superheating, of condensation, of the ratio of expansion and the economy of the engine.

We hope that this last machine of 150 indicated horsepower will find grace in the eyes of Professor Heck and will not be, like the two others and that of the laboratory of Liège, considered as too small. In any case it differs sufficiently from those of Donkin and of Willans that one can deduce by experiment the certitude that the size of the machine has not a sensible effect on the disturbances brought about by the exchange of heat between the metal of the cylinder and the working vapor.

Since I have indicated my sources of information it is possible for the reader to verify the conclusions of the tests in question, and of which a table will be found at the end of this article. I have abstained from presenting the data of the tests and have given simply the calculated results, on the subject of which a little preliminary explanation is due.

Contrary to the ordinary theory of the steam engine, the only one which Professor Heck apparently sees, is what experiment has revealed on the action of the heat carried by the steam into the cylinders of our engines.

During admission a part of the inflowing steam condenses against the walls, chilled on the preceding stroke by communication with the condenser.

During the expansion a part of this water condensed on the surface of the walls during admission is reëvaporated usefully, contributing to increase the exterior work. But ordinarily the water is not entirely evaporated, and there remains at the end of the expansion and at the moment when communication with the condenser is established some still unevaporated.

During exhaust the rest of the water which was on the walls is evaporated simply to pass through the condenser, which constitutes a complete loss so far as work is concerned. For one readily sees that this water thus coming to the condenser has come in the form of steam from the boiler, has passed through the cylinder by the favor of the metal, and has entered the condenser anew under the form of vapor, without having in any fashion contributed to the work done.

We will not talk of the phase of compression in the clearance space in order not to unnecessarily complicate the argument, and, in the same spirit, we will neglect the steam and exhaust leads, as well as the small quantity of steam which

fills the clearance at the end of the exhaust.

Therefore, during the admission there comes from the boiler into the cylinder a weight M_a pounds of steam, which separates itself into two distinct parts: the one in the gaseous state occupying a volume V_o corresponding to the pressure p_o indicated upon the diagram, of which we will call m_o the weight; the other, in the liquid state, is spread over the surface of the walls, and its weight is $M - m_o$.

The difference $M_a - m_o$ has received the name of missing quantity at the end of admission, and for this reason: In order to estimate m_o, the weight of the saturated steam present in the cylinder at the end of admission, and occupying the volume V_o corresponding to the pressure p_o, one finds in the steam tables the weight d_o pounds per cubic foot of saturated steam, and the product $v_o\ d_o$ is equal to m_o. As to M_a, it is a quantity determined directly by experiment. The ratio

$$\frac{m_o}{M_a} = x_o$$

is called in French *"titre du mélange,"* and in English "quality of the steam."

It is this which it is necessary to know for the discussion of engine economy. This is also true of the ratio

$$\frac{M_a - m_o}{M_a} = 1 - x_o$$

It should not be concluded that the missing quantity is of small importance. In the tests recorded in the final table this quantity $1 - x_o$ varies between 20 and 44 per cent.

In the same way at the end of the expansion there remains in the cylinder a volume v_1 corresponding to saturated steam of the pressure p_1 of which the weight is $V_1\ d_1 = m_1$ pounds. Its quality is, therefore, $x_1 = \frac{m_1}{M_a}$ and the missing quantity equals

$$1 - x_1.$$

Generally during the expansion a part of the $(M_a - m_o)$ pounds of water is evaporated, with the result that one has

$$x_1 > x_o \text{ and } 1 - x_1 < 1 - x_o$$

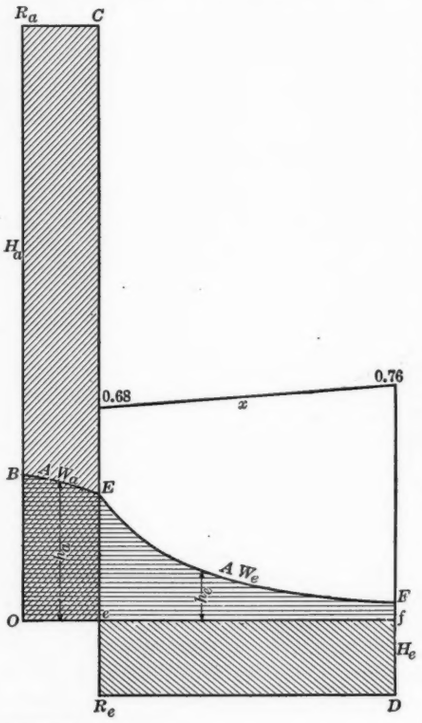
For the eight tests of engines recorded in the final table the values of $1 - x_o$ and of $1 - x_1$ are given. Let us now pass to the valuation of the quantities of heat in play.

The M_a pounds of steam coming into the cylinder for one stroke of the piston bring in Q thermal units, of which a part disappear to produce the work W_a foot-pounds. This part equals $\frac{W_a}{778}$ B.t.u., which we will call $A\,W_a$, representing the reciprocal of 778 by A. A second part, R_a thermal units, represents the heat given up to the metal of the cylinder walls. The rest

$$V_o = Q - A\,W_a - R_a$$

is present in the steam at the commencement of expansion. The experimental theory gives the means to calculate R_a by this equation, in furnishing experimentally the value of V_o, of Q and of $A\,W_a$.

During the expansion the steam which had V_o thermal units at first loses $A\,W_e$ thermal units to produce the work of expansion. It gains the heat R_e thermal units that the walls restore in vaporizing a part of the water which covers them and finishes by still containing U_1 thermal



units from now on completely lost, so that

$$U_o - A\,W_e + R_e = U_1$$

Experiments giving the values of U, $A\,W_e$ and U_1 make it possible to deduce the value of R_e from this last equation.

To recapitulate, for the entire stroke of the piston, the heat utilized in work is

$$A\,W_a + A\,W_e = A\,W_f$$

The loss to the cylinder walls,

$$R_a - R_e = R_f$$

Loss by heat of exhaust steam, U_1.

The Diagram of Heat Exchange

Whatever the length of the stroke of the piston in the engine in question, that stroke is represented invariably upon the diagram by a length of two inches $= O\,f$ in the accompanying figure. The fraction of the stroke passed through during admission is represented by $O\,e$, and that during expansion by $e\,f$. The steam line $B\,E$ and the expansion line $E\,F$ of the indicator diagram are traced in upon a convenient scale. Then the area $O\,B\,E\,e\,O$ represents upon a certain scale the heat equivalent of the work performed by W pounds of steam during admission; that is to say, $\frac{W_a}{778}$ B.t.u., that we call $A\,W_a$ in making $A = \frac{1}{778}$.

In the same way and to the same scale the surface $e\,E\,F\,f\,e$ represents the heat equivalent of the work furnished by the steam during expansion. In order to distinguish these areas I have covered them with horizontal rulings.

We will call h_a and h_e respectively the hights of two rectangles, of which the bases will be $O\,e$ and $e\,f$, and of which the surfaces are $A\,W_a$ and $A\,W_e$.

Now, knowing by experiment the quantities of heat R_a furnished by the steam to the metal during admission, and R_e restored by the metal to the steam during expansion, these may be represented by surfaces on the same scale as those proportional to the work effected. In order to represent R_a thermal units a rectangle is drawn, of which the base is $O\,e$ and the hight H_a, calculated from the equation

$$O\,e \times H_a = R_a$$

that is, the rectangle $O\,B\,R_a\,C\,E\,e\,O$. In the same way we proceed to represent R_e, which gives a rectangle having $e\,f$ for the base and H_e for hight, calculated by the equation

$$e\,f \times H_e = R_e$$

The sign for R_a is, however, the contrary of the sign for R_e, the one representing the heat ceded by the steam to the metal, and the other by the metal to the steam. For this reason we place the first rectangle R_a above the axis $O\,f$ and the second below, distinguishing the surfaces besides by different inclinations of the cross-hatching.

The diagram of heat exchange is, therefore, $O\,B\,R_a\,C\,E\,e\,R_e\,D\,f\,O$. It is easy to trace it if one knows the ratio of R_a to $A\,W_a$ and that of R_e to $A\,W_e$, ratios equal to $\frac{x\,H_a}{h_a}$ and $\frac{H_e}{h_e}$ respectively.

We add to the figure a line of which the ordinates represent the quality x of the steam during the expansion, and of which the values are $x_o = 0.68$ at the commencement and $x_1 = 0.76$ at the end. We suppose, although this will not be exact, that the diagram of this quality will be a straight line. The diagram is drawn upon the scale of two inches equals unity. Under the conditions assumed, the heat R_a lost during admission is 4.47 times greater than the heat $A\,W_a$ utilized for the work, and it is greater

than the heat R_e restored by the metal during exhaust, with the result that there is a positive loss equal to

$$R_a - R_e = R_f$$

which we will call the final loss R_f. It is important to consider also the ratio of the final loss R_f to the heat equivalent of the final work

$$A\,W_f = A\,W_a + A\,W_e$$

In the accompanying table we give these ratios

$$\frac{R_a}{A\,W_a},\quad \frac{R_e}{A\,W_e},\quad \frac{R_f}{A\,W_f}$$

which permit diagrams of the heat exchange for the eight tests of Donkins, Willans and Hirn to be traced. One who has not studied in actual operation the evolutions of steam in the cylinder will perhaps with difficulty believe that the quantity of heat ceded to the metal during the admission can amount to 4.47 times that which represents the work effected during the same period. He will believe without doubt that our diagram is exaggerated, but if he will cast his eyes over our table of experimental data he will see that the figure 4.47 was obtained in text X of the Willans engine at 400 revolutions per minute, and that at 138 the same ratio has increased to 9.20, that in the VI test of the Hirn engine of 150 horsepower this ratio amounted to 6.6, while in the two tests II and IV on the Donkin engine of six horsepower the ratios were 3.55 and 2.67 respectively. In fact, our diagram is very approximately that of test X of Willans as to degree of expansion, as to work diagram, diagrams of exchanges and to qualities x_0 and x_1. Our diagram makes very apparent the relative importance of the heat exchanges between the metal and the steam comparatively to the heat converted into work. It presents this advantage over the entropy diagram, which conceals rather than illustrates the real phenomena and which is plotted only with great difficulty and by graphic calculations impossible to control, while our diagram is based on figures obtained by arithmetical calculations. But let us return to the principal object of our discussion and commence by giving the table of tests chosen as enumerated above.

Is Professor Heck justified in believing that the disturbance brought about by the thermal action of the walls are sensible in small engines and negligible in the larger machines, such as one finds in practice? The accompanying table gives the results obtained on three machines, one large of 150 horsepower, another small of 6 horsepower, the third intermediate of 35 horsepower. It seems to me that the differences are sufficient to show themselves. Well, the examination of the table leads to the following conclusion. The loss by the walls during admission varies according to circumstances from 4.33 to 6.59 in the large machine, 2.67 to 3.55 in the small machine, 4.47 to 9.20 in the intermediate machine.

The restitution by the walls during expansion from 0.80 to 1.22 in the large engine, 1.90 to 2.50 in the small engine, 1.20 to 1.90 in the intermediate engine.

The positive loss by the walls from 0.97 to 1.94 in the large engine, 1.51 to 1.70 in the small engine, 1.05 to 2.30 in the intermediate engine.

The proportion of steam condensed during admission varies from 25 to 37 per cent. in the large engine, 20 to 23 per cent. in the small engine, 32 to 44 per cent. in the intermediate engine.

The proportion of water reëvaporated during expansion varies from 2 to 6 per cent. in the large engine, 3 to 4 per cent. in the small engine, 8 to 10 per cent. in the intermediate engine.

To sum up, the action of the cylinder walls does not vary with the size of the engine, but considerably with the conditions of operation. If this action is more intense in the intermediate machine, it is that this was single-acting, and its walls exposed for one-half of the time to free and cool air.

DATA FROM TESTS OF DONKIN, WILLANS AND HIRN.

Object of the Tests.	Number.	Condensation.	Degree of Expansion.	Revolutions per Minute.	Experimental H.P.		Pounds of Steam per Hour.		Missing Quantity.		Ratio of the Exchange of Heat R to the Heat Equivalent of the Work.		
					Absolute.	Indicated.	Per Absolute H.P.	Per Indicated H.P.	At Beginning of Expansion. $1-x_0$	At End of Expansion. $1-x_1$	$\frac{Ra}{AWa}$ During Admission.	$\frac{Re}{AWe}$ During Expansion.	$\frac{Rf}{AWf}$ During both Admission and Expansion.
Donkin 6 H.P. Effects of condensation	II.	Condens.	0.684	92.27	8.03	5.61	26.14	37.41	0.23	0.19	3.55	2.5	1.70
	IV.	No Cond.	0.684	92.66	11.74	6.26	22.74	42.65	0.20	0.17	2.67	1.9	1.51
Willans 35 H.P. Effects of speed	X.	No Cond.	0.28	400.16	49.33	35.33	17.90	24.75	0.32	0.24	4.47	1.2	1.05
	XI.	No Cond.	0.28	223.7	28.27	20.58	22.00	30.06	0.44	0.32	8.41	1.8	1.95
	XII.	No Cond.	0.28	138.0	17.97	13.36	22.90	30.06	0.44	0.34	9.20	1.9	2.30
Hirn 150 H.P. Effect of superheat and of ratio of expansion	II.	Condens.	0.257	30	160.64	136.46	18.27	21.51	0.31	0.25	4.52	1.2	1.26
	V.	Condens.	0.162	30	125.20	113.08	14.59	16.16	0.25	0.22	4.33	0.9	0.97
	VI.	Condens.	0.162	30	121.75	107.31	17.18	19.41	0.37	0.35	6.59	0.8	1.94

Rejoinder by Prof. R. C. Heck

In so far as the foregoing is controversial in form, it is to a large degree based upon misconception, as has been already pointed out. Assumptions and "theories" are attributed without any real foundation in my article of September 13. It is regrettable that the tone of personality should be so strongly injected into a presentation of scientific information, and so much space wasted on the demolition of theories which are now held by no intelligent thinker.

The thesis still stands that in its combination of small size and low speed the engine in the laboratory at Liège is not representative of commercial practice, and that deductions from the results of experiments upon it cannot be directly applied to large or fast-running engines. The same actions exist in large engines, but their relative magnitude is less.

In the foregoing paper the discussion which follows the presentation of references and data, describing the general action of the cylinder walls in causing condensation and reëvaporation and in wasting heat on the exhaust steam, is all common knowledge; no one questions it or is ignorant of it. The scheme of graphical representation of thermal action is one of the recognized methods, although it is doubtful whether engineers in this country ever use it. For most purposes, and especially when considering such an involved and uncertain matter as the way in which the action of the cylinder walls varies in different engines and under different conditions, the mere percentages of steam initially condensed and then reëvaporated constitute

a close enough measure of the wall effect.

The Hirn engine which is quoted certainly had a large cylinder, but was of slow speed. However, in the sense of output it was not a large engine; an engine, excluding pumping engines, only begins to be considered large at 500 horsepower. That the Donkin engine shows less condensation than the Hirn engine is accounted for by higher speed and late cutoff, which factors overbalance the smallness of the cylinder. The Willans engine has its marked peculiarities, and besides is no bigger (in piston displacement) than the Donkin engine; as a minor correction, the size is 14x6 inches, not 13x6 inches. Altogether, the data presented are too few and too discordant to give a clear idea of the influence which any of the controlling conditions exert upon cylinder action. These governing conditions are, speed in revolutions per minute (not piston speed), size (with which the type of cylinder design must be included), ratio of cutoff or of expansion, and range of pressure and steam temperature within the cylinder.

Reciprocating Engine and Low Pressure Turbine*

Some interesting figures in support of the low-pressure turbine as used in connection with the reciprocating engine were shown by the tests of the steam yacht "Vanadis." This vessel, which is of 1300 tons displacement and 279 feet length overall, was built in 1908 and fitted with three Parsons turbines, one high-pressure and two low-pressure. The builders guaranteed a coal consumption of 26 tons per 24 hours when cruising at 13 knots, but it was found that in actual service this figure was greatly exceeded, in fact, so much so that the steaming radius with the limited bunker capacity was cut unconveniently short.

To remedy this, it was decided to remove the high-pressure turbine and replace it by a triple-expansion reciprocating engine, leaving the low-pressure turbines connected to the outboard shafts. After the completion of these changes a set of standardization trials were made. First, the propellers were removed from the turbine shaft and the vessel run at 13 knots with the reciprocating engine alone, during which the steam consumption was approximately 17 pounds per indicated horsepower-hour. Next, the propellers were replaced and the reciprocating engine run in connection with the two turbines—a speed of 13 knots being maintained—in which case a water consumption of 14½ pounds per indicated horsepower-hour was attained, as against 20½ pounds before the change was made.

*Abstracted from a paper read by C. H. Crane before the Society of Naval Architects and Marine Engineers.

A Rule of Thumb for Horsepower

By F. R. Low

The horsepower of an engine is the product of the piston area, the piston speed and the mean effective pressure divided by 33,000.

The piston area is 0.7854 times the square of the diameter.

The complete formula then is;

$$H.P. = \frac{0.7854\ D^2 \times \textit{piston speed} \times M.E.P.}{33{,}000}$$

Dividing the 33,000 by the 0.7854 this becomes,

$$H.P. = D^2 \times \frac{\textit{piston speed} \times M.E.P.}{42{,}017}$$

The quantity by which the square of the diameter is to be multiplied will be for the usual case somewhere between 0.5 and unity. When it is unity, i.e., when the product of the piston speed and the mean effective pressure is 42,017, the formula becomes delightfully simple

$$H.P. = D^2.$$

This would be true of any of the combinations of piston speed and mean effective pressure given in the first double column of the accompanying table. A condensing engine might easily have a mean effective pressure of 52.5 pounds and run at 800 feet piston speed, and for such an engine this simple formula would give out of hand an excellent idea of its capacity.

The remaining double columns of the table give the combinations of piston speed and mean effective pressure which would justify the use of the single-place numbers 0.9, 0.8, 0.7, etc., at the heads of the column. The common assumption of 600 feet of piston speed and 40 pounds mean effective pressure would call for 0.57 to which a column is devoted, but 0.6 D^2 would give a close approximation to this condition.

The idea is that the horsepower will usually lie between

$$H.P. = D^2$$

and

$$H.P. = 0.5\ D^2 \text{ or } \frac{D^2}{2}$$

For simple condensing engines at high piston speeds the first and simpler formula will give a close approximation. With lower piston speeds and mean effective pressures the square of the diameter may have to be multiplied by a factor running down to 0.5 for the conditions given in the last column of the table.

$H.P. = D^2$ $P. \times S. = 42{,}017$		$H.P. = 0.9\ D^2$ $P. \times S. = 37{,}815$		$H.P. = 0.8\ D^2$ $P. \times S. = 33{,}614$		$H.P. = 0.7\ D^2$ $P. \times S. = 29{,}412$		$H.P. = 0.6\ D^2$ $P. \times S. = 25{,}210$		$H.P. = 0.57\ D \times \frac{D^2}{1.75}$ $P. \times S. = 24{,}000$		$H.P. = 0.5\ D^2 = \frac{D^2}{2}$ $P. \times S. = 21{,}008$	
Piston Speed.	M. E. P.	Piston Speed.	M. E. P.	Piston Speed.	M. E. P.	Piston Speed.	M. E. P.	Piston Speed.	M. E. P.	Piston Speed.	M. E. P.	Piston Speed.	M. E. P.
300	140.1	300	126.0	300	112.0	300	98.0	300	84.0	300	80.0	300	70.0
350	120.0	350	108.0	350	96.0	350	84.0	350	72.0	350	68.6	350	60.0
400	105.0	400	94.5	400	84.0	400	73.5	400	63.0	400	60.0	400	52.5
450	93.4	450	84.0	450	74.7	450	65.4	450	56.0	450	53.3	450	46.7
500	84.0	500	75.6	500	67.2	500	58.8	500	50.4	500	48.0	500	42.0
550	76.3	550	68.7	550	61.1	550	53.5	550	45.8	550	43.6	550	38.2
600	68.7	600	63.0	600	56.0	600	49.0	600	42.0	600	40.0	600	35.0
650	64.6	650	58.2	650	51.6	650	45.2	650	38.8	650	36.9	650	32.3
700	60.0	700	54.0	700	48.0	700	42.0	700	36.0	700	34.2	700	30.0
750	56.0	750	50.4	750	44.8	750	39.2	750	33.6	750	32.0	750	28.0
800	52.5	800	47.3	800	42.0	800	36.8	800	31.5	800	30.0	800	26.2
850	49.4	850	44.4	850	39.5	850	34.6	850	29.6	850	28.2	850	24.7
900	46.7	900	42.0	900	37.3	900	32.7	900	28.0	900	26.6	900	23.3
950	44.2	950	39.8	950	35.4	950	31.0	950	26.5	950	25.3	950	22.1
1000	42.0	1000	37.8	1000	33.6	1000	29.4	1000	25.2	1000	24.0	1000	21.0
1050	40.0	1050	36.0	1050	32.0	1050	28.0	1050	24.0	1050	22.8	1050	20.0
1100	38.2	1100	34.4	1100	30.5	1100	26.7	1100	22.9	1100	21.8	1100	19.0
1150	36.5	1150	32.9	1150	29.2	1150	25.6	1150	21.9	1150	20.9	1150	18.3
1200	35.0	1200	31.5	1200	28.0	1200	24.5	1200	21.0	1200	20.0	1200	17.5

On account of numerous cases of cholera, which it is thought may be traced to that source, the Minister of the Interior of Hungary has forbidden the cutting of ice from ponds and rivers. This would seem to open an unusual opportunity for builders of ice and refrigerating machinery in that country.

A good telltale that will show when a bucket trap is not working or is getting more water than it has capacity to handle can be made by connecting a brass-tube air valve to the top of the trap, which will blow whenever the trap is full of water.

Gas Power Department

The Gas Cleansing Plant at the Lackawanna Steel Works*

By E. P. Coleman

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men

At the Buffalo works of the Lackawanna Steel Company is located the first gas-engine power plant to be operated in this country with blast-furnace gas. As far as can be learned, the selection of the type of engine was made in 1900, based on extended observation by a committee of the working of blast-furnace gas-power plants in Europe. The types observed were the Cockerill, Otto, Oechelhaueser and Koerting. The Oechelhaueser engine was disregarded on account of the crank-shaft design and the four-stroke cycle engines were not favorably considered on account of exhaust-valve troubles which did not seem to have been mastered at that time. The engines were built by the De La Vergne Machine Company, New York, after designs by the firm of Koerting Brothers, Hanover, Germany.

At this plant 12,000 to 15,000 net horsepower is normally developed by gas engines operated with blast-furnace gas, the greater portion for blowing the furnaces. There are sixteen blowing engines, each rated at 2000 indicated horsepower, and eight electric power units, each consisting of a gas engine rated at 1000 indicated horsepower, direct connected to a 500-kilowatt generator. Four of the latter units generate direct current at 250 volts and the other four generate three-phase 25-cycle alternating currents at 440 volts. All of the engines are of the two-stroke cycle type, equipped with twin cylinders and cranks spaced 90 degrees apart.†

The general arrangement is shown in Fig. 2. There are six blast furnaces in a line extending approximately north and south. These furnaces are grouped in pairs, each pair forming a unit with reference to the arrangement of its stoves, ore bins, gas plant and various auxiliaries. The engines in blowing-engine house No. 2 furnish air for furnaces 3 and 4. Furnaces 5 and 6 are supplied with air by the engines in blowing-engine house No. 3. The air for furnaces 1 and 2 is usually supplied by steam engines located in the north end of blowing-engine house No. 2. The gas-driven electric generator units are located in the south end of power house No. 1.

The general process of preparing the gas for use in the engine cylinders is as follows: The gas leaving the furnace top passes through large downcomer pipes to the dust catcher, where the heavier portion of the dirt is deposited under the action of gravity. From the dust catcher, portions of the gas pass through pipe mains respectively to the hot stoves, the boilers and the gas engines. That portion used by the gas engines is cooled and partially cleaned by means of water sprays in the pipes and chambers through which the gas passes; after which it is further cleaned by passing through centrifugal fans into which water is also sprayed. From the fans the gas passes through separators which remove the entrained moisture and its entrained dirt, and thence under a few ounces of pressure (above atmosphere) to the engine houses.

The six blast furnaces under consideration produce about 2600 long tons of iron per 24 hours, or about 108 tons per hour. The gas amounts to about 150,000 cubic feet per ton of iron, or, say, about 16,000,000 cubic feet per hour or more from the six furnaces. Approximately 2,500,000 cubic feet per hour is used for the gas-engine plant, the remainder being burned in the hot-blast stoves and under the boilers.

All of the dust catchers are provided with a suspended partition or baffle wall of firebrick cutting off direct passage of gas from inlet to outlet; the gas having to pass under this wall and up to the outlet.

Piping

The general arrangement and main dimensions of the piping for washed gas are shown in Fig. 1. Each group of eight blowing engines is served with washed gas through a 60-inch overhead main of riveted steel plate delivering gas to a main header of 96 inches diameter alongside the wall of the engine house near the yard level. The eight 1000-horsepower engines at power house No. 1 are supplied with gas from furnaces 1 and 2 through a 30-inch underground pipe of cast iron. This pipe is cross-connected with the 96-inch header near the south end of blowing-engine house No. 2, by means of a 42-inch overhead pipe. Gas washers 3 and 4 are connected with gas washers 5 and 6 through a 36-inch equalizing pipe, as shown. Low points in the piping are provided with drains. The general arrangement of this piping deserves adverse criticism, as will be shown.

No gas holders are installed or required. The length of the 30-inch main supplying the power house is about 1180 feet. The 60-inch main supplying engine house No. 2 is about 525 feet long, and the 96-inch header at blowing-engine house No. 3 is supplied through about 415 feet of 60-inch pipe. In this main

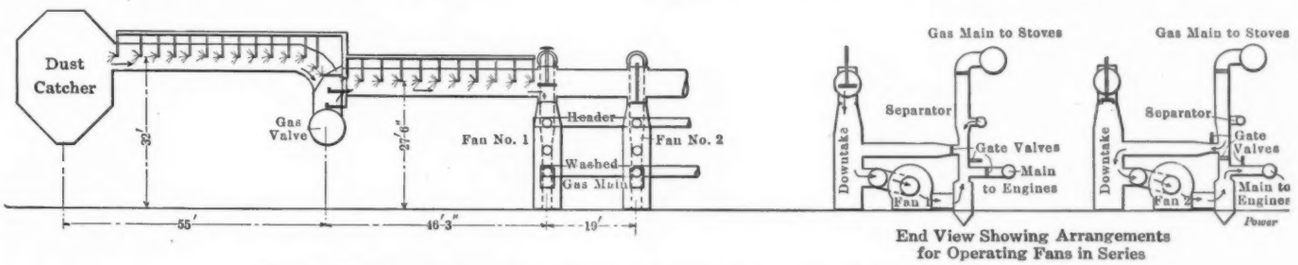


Fig. 1. Schematic Diagram of Gas-washing Plant for Furnaces 1 and 2

*Partial abstract of a paper read before the American Society of Mechanical Engineers, December, 1910.

†These engines were described in Power several years ago; hence a detailed description of them is omitted here.—Ed.

at the south end of the engine house is a 60-inch venturi meter. The blowing-engine house header lies along the east side of the building. It is supported on the concrete work of the exhaust tunnel and is about 400 feet long. The 8-foot section is 255 feet long and is of riveted ⅜-inch plate. The plates of the 6-foot portion are 5/16 inch thick. A 24-inch connection is taken off for each engine on the side nearest the building. Water is drained from the header by means of an inverted siphon. The total length of piping from the washers to the engine houses is about 3800 feet.

Nos. 1 and 2 Gas Washers

There is a gas-cleaning plant at each pair of furnaces consisting of chambers equipped with water sprays for cooling the gas and washing out a portion of the dirt, centrifugal fans also provided with water sprays, separators for removing the entrained water, and the necessary valves and piping. Schematic diagrams, Figs. 1 and 3, show the general arrangement of the washing plants, the former illustrating the washery first installed at furnaces 1 and 2, and Fig. 3 the arrangement of apparatus at furnaces 3 and 4. The general arrangement of washers at furnaces 5 and 6 is similar to that at furnaces 3 and 4. In the plant represented in Fig. 1 the gas is taken from the dust catchers through horizontal pipes where it is given an initial cooling and washing by means of water sprays. The cool gas then passes to four fan washers located between the two furnaces. These fans are normally operated in pairs, each pair forming a unit consisting of the two fans operating in series with each other; the first fan draws cool gas from the main supply and discharges it to a first-washed main, and the second fan takes its gas from this main and discharges it to the second-washed main, from which the gas passes to the 30-inch gas line and to the power house.

The pipe leading from the dust catcher to the fans is 70 inches in diameter. At a point about midway between the fans and the dust catcher a water-seal valve is located, consisting of a horizontal steel tank 8 feet in diameter, through which the gas passes on its way to the fans. By filling with water it acts as a shut-off valve and by partly filling with water the gas flow may be reduced to any desired extent, these functions being useful when a furnace is working badly and giving a poor quality of gas. The 8-foot tank also serves as a receptacle to which the water is drained from the cooling sprays.

The 70-inch pipe connecting the dust catchers to the fans is about 236 feet long, the total travel of the gas from dust catcher to fan being about 123 feet. Located in this piping are 99 water sprays for cooling the gas on its way to the fan washers. These spray nozzles are located at the axis of the pipe, about 3 feet 6 inches apart, and discharge a cone-shaped spray into and against the stream of gas. The sprays are supplied from a 3-inch header through 1-inch pipe connections and each spray consumes about 8 gallons of water per minute. The water drains first into the 8-foot tanks and from thence through a seal into tank cars beneath, where the dirt is deposited and the water passes off from an overflow into the general drainage system.

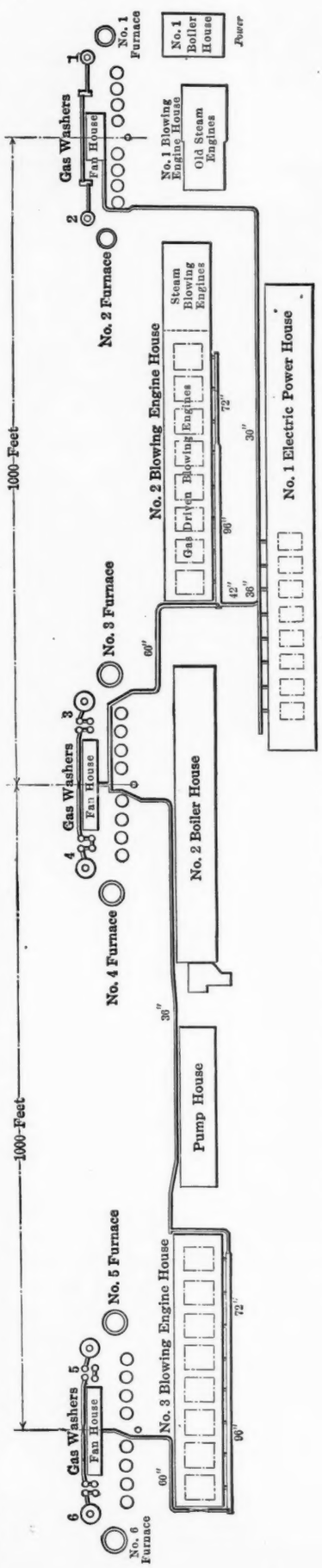


Fig. 2. Plan Diagram of Furnaces, Washers and Power Houses of Lackawanna Steel Works

The connections to the fans are taken from the bottom of the 70-inch main, the connection being 48 inches in diameter, enlarging to 70 inches diameter. Each connection can be shut off from the gas pipe by means of a disk valve operated by a chain drum and handwheel located on top of the gas pipe, and each connection is provided with a hopper bottom, valve and drain, forming a pocket for the mud and water brought down with the gas. The drain pipe extends downward into a well, forming a seal.

The fans are very similar in general features to ordinary centrifugal ventilating fans. The wheels are 6 feet 11 inches in diameter. There are 8 blades, each 15½ inches wide at the inner end and 13 inches wide at the tip, carried on tee-iron arms set in a cast-iron hub. The cast-iron suction connections are rectangular. The main is 20¼x52 inches. A branch 21x48 inches leads to each side of the fan, the opening to the fan casing being 36 inches in diameter. These connections are provided with cleaning holes to facilitate removal of mud. Water connections are provided for four nozzles on each side carried through the casing and discharging through the circular inlet to the fan. Waste water from the furnace tuyeres and bosh plates flows from the furnace troughs into a stand pipe equipped with an overflow located at the proper level, and a portion of the water in the stand pipe passes through pipes to the fans.

Each fan is driven by a 75-horsepower electric motor direct connected to the fan by means of a flexible coupling, and each one discharges horizontally at the bottom through a 21½x45-inch connection into a water separator. The separator is a steel box 4 feet square by 9 feet high, containing a set of baffles consisting of three rows of 3-inch steel channels, the flanges of the channel bars facing the stream. The openings between channels are about 1 inch wide, and the spacing is alternate or staggered, such that the streams of gas are broken and turned. The separated water and mud drop to the bottom of the separator and pass out through a seal. The gas leaves the separator at the top through a 24-inch pipe connection.

First-washed gas which has passed through one fan passes back into one of the 70-inch vertical connections on the cool gas main, and is isolated from it by

means of the disk valve at the top, previously mentioned. The gas then flows through the second-wash fan to the second-washed gas main; and thence through the 30-inch line to the power house. The piping and valves are so arranged that any fan may be used for either first or second washing. The valves in the fan connections are 24-inch gates, with seats and disks of cast iron.

Nos. 3 and 4 Gas Washers

In the washing plant shown in Fig. 3, the gas passes from the dry-dust catcher through a 96-inch connection leading to a set of four cooling towers 12 feet in diameter and 72 feet high. The cooling water is sprayed into these towers through numerous nozzles set in the sides. The first-washed gas passes from the fans through a water separator into a 78-inch header called the first-wash main, from which it is passed back to the suction side of the fans working on second washing, these being shut off from the cooled-gas main. From these fans the gas passes through the separators and into the second-washed main of 60 inches diameter. The 60-inch main supplying the engines is connected to the middle point of this header.

The four cooling towers are carried on a structural platform which is located 18½ feet above yard level to provide clearance for mud cars which receive the drainage from the bottom connections. The connections between the towers are 8 feet in diameter and the travel of the gas is up and down in alternate towers. Each tower is provided with a hopper bottom and pipe seal having connection with a common drain pipe which delivers the discharge of all four towers to the tank cars beneath. Towers *A* and *B* are further provided with emergency seal drains at a higher level. These emergency seals are normally idle, but should the bottom drain become clogged the tower will then drain through the emergency seal. Tower *A* is also provided with a steel plate baffle in front of the inlet opening. When the tower is filled with water to the required hight the water and baffle serve as a valve to shut off communication between the towers and the dry-dust catcher, a sealed overflow maintaining the proper water level when desired.

The first tower is equipped with about 30 sprays, the second has 35, and the third and fourth about 20 to 25 sprays each. The sprays are placed in five circular rows, 6 feet apart vertically; the lower row is about 20 feet from base of tower. The nozzles are of brass throughout, and made as shown in Fig. 4. The shell has a 2½-inch external pipe thread which screws into a flange riveted to the shell of the tower. The helical passage produces a whirling cone-shaped spray of about 90 degrees "spread" and 6 to 10 feet maximum diameter. The 1½-inch plug permits access to the spindle for cleaning. These sprays use about 7 gallons per minute at the average pressure carried.

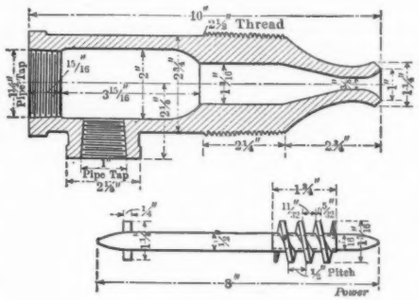


Fig. 4. Spray Nozzle

From the towers the gas passes through a 7-foot pipe to the fans. This main by means of bell valves operated by a winch and handwheels located on top of the horizontal main. These valves seat in a water seal. The opening is of 45 inches diameter. At the midpoint of the cooled-gas main connecting the two sets of towers is located a shut-off valve consisting of an inverted siphon which may be partly or wholly filled with water to regulate the amount of gas coming from either furnace.

There are eight fan washers housed in a steel-frame building and set with their shaft centers 9 feet 5 inches above the yard level, this elevation being necessary for drainage. The wheels are of ¼-inch steel plate, 7 feet 1 inch in diameter, having eight blades each 29½ inches wide at the inner end, and 27½ inches wide at the tip. The central opening of wheel is 40 inches in diameter. The casing is of cast iron and of the double-suction type; each branch of the suction connection is rectangular, 20x50 inches, opening into the fan through a 36-inch diameter inlet. The bottom-discharge connection is 40x36 inches. The wheel shaft is direct connected through a flexible coupling to a 100-horsepower motor running at 480 revolutions per minute. Six of these motors utilize alternating current and two direct current. The two types are used so that should there be an accident to either circuit, the motors on the remaining circuit will continue to operate and keep a part of the engines running until the necessary corrections can be made.

Water is thrown into each fan through 16 spray nozzles. Eight of these discharge into the central inlet openings, four on each side; the other eight discharge into the fan through the upper half of its periphery. The sprays are fed by four 1½-inch lines connected to a 6-inch header extending the full length of the building, over the fans. Each spray connection is a ¾-inch pipe. The water for the fan sprays is taken from the waste of bosh-cooling water from the furnace, which overflows from a standpipe, a portion under the necessary head going to the fans.

From the fan washers the gas passes to water separators constructed as shown

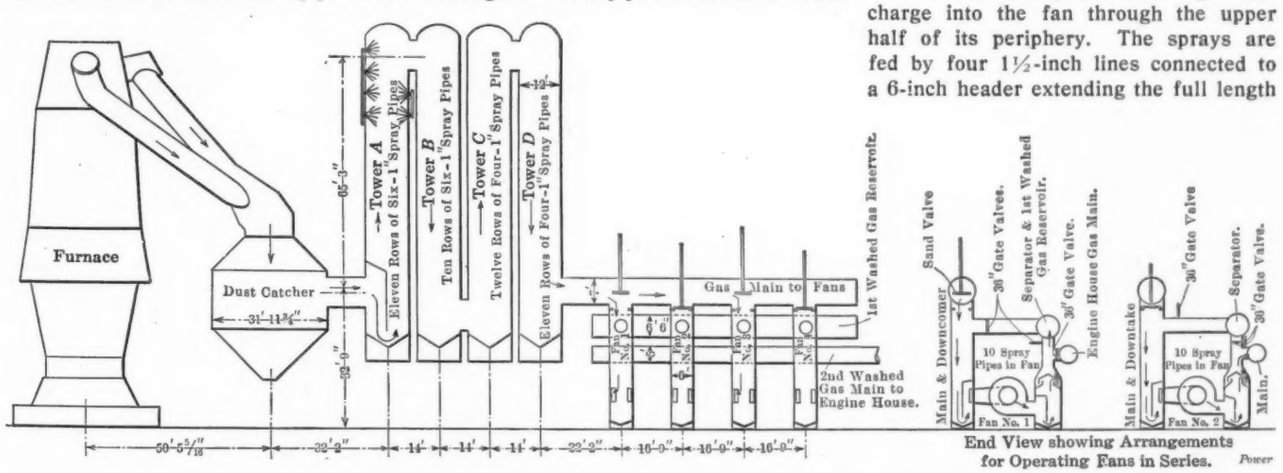


Fig. 3. Schematic Diagram of Gas-washing Plant for Furnaces 3 and 4

in Fig. 5. The gas passes first through a set of baffles consisting of four rows of 4-inch channels set vertically, the openings between the channels being 1½ inches wide, with staggered spacing so that the streams of gas are broken and turned. The separated water falls down the vertical channels, carrying with it the dirt, and passes out at the bottom through a seal to the drainage system. After passing through the channel baffle, the gas rises through annular disk baffles and passes from the separator through a 36-inch top connection.

Each separator is equipped at the top with a cast-iron tee providing outlets through 36-inch gate valves to the first-washed and second-washed mains. Fans working on first washing discharge their gas into the first-washed main. This gas is then taken by the fans working on second washing and by them discharged into the second-washed main. The diameters of the first- and second-washed mains are 78 inches and 60 inches. The first-washed main extends the full length of the fan house, and is connected to the vertical suction connection of each fan through a 42-inch gate valve. The valves and piping are so arranged that any fan may be operated on either first or second washing.

The gas-washing plant at furnaces 5 and 6 is substantially the same as that described for furnaces 3 and 4, but the water separators have a different style of baffling. One of these separators is shown in Fig. 6. The gas circulates through zigzag passages formed by narrow plates assembled as shown. The projecting edges of plates are formed to catch the water and lead it to the bottom of the chamber, where it passes out through a seal. There are two sets of baffles through which the gas passes in succession, one at the bottom and one at the top.

General

The delivery mains from the three gas-cleaning plants are interconnected by two pressure-equalizing pipes. These mains for hot gas, cooled gas and washed gas are locally interconnected in parallel relation at each pair of furnaces. It is therefore possible to control the amount of gas taken from each furnace and the gases from the two furnaces are thoroughly mixed by discharging into a common washed-gas delivery main. In order to promote in the best manner uniformity in the composition of the gas, the joint delivery from the several washeries should then discharge into a common distributing main or holder. The locations of the three delivery mains, however, and the relative locations of the two equalizing pipes are such that it is impossible for such mixing to occur even locally or approximately, as reference to Fig.1 will make clear.

A partial solution constituting a great improvement would consist in relocating the 30-inch delivery pipe from washers Nos. 1 and 2 along the west wall of blowing house No. 2, to form a junction at the southwest corner of that building with the 60-inch delivery main from washers Nos. 3 and 4. Also the 36-inch equalizing main should deliver gas into the 60-inch delivery main from washers Nos. 5 and 6, instead of into the north end of the 96-inch header at blowing house No. 3. Power house No. 1 and blowing house No. 2 would then receive the average of gas from four furnaces, whereas at present blowing house No. 2 receives gas only from furnaces 3 and 4, and at power house No. 1, the four north engines may receive gas from furnaces 1 and 2, and the four south engines from furnaces 3 and 4. Under these conditions of piping, the gas supply at any point is but an average of that from two furnaces, and at times the irregularity is considerable, the heat value occasionally varying between the limits of 105 and 80 B.t.u. per cubic foot, within a period of a few seconds.

Both the gas-cleaning apparatus and the gas engines were installed at an early stage in the history of the art and are necessarily imperfect when compared with modern examples to which have been applied those refinements that can be gained only through experience. The average dust content of the gas delivered to the second-washed main amounts to about 0.022 to 0.035 grain per cubic foot, which would rightly be considered bad practice in modern gas-cleaning plants.

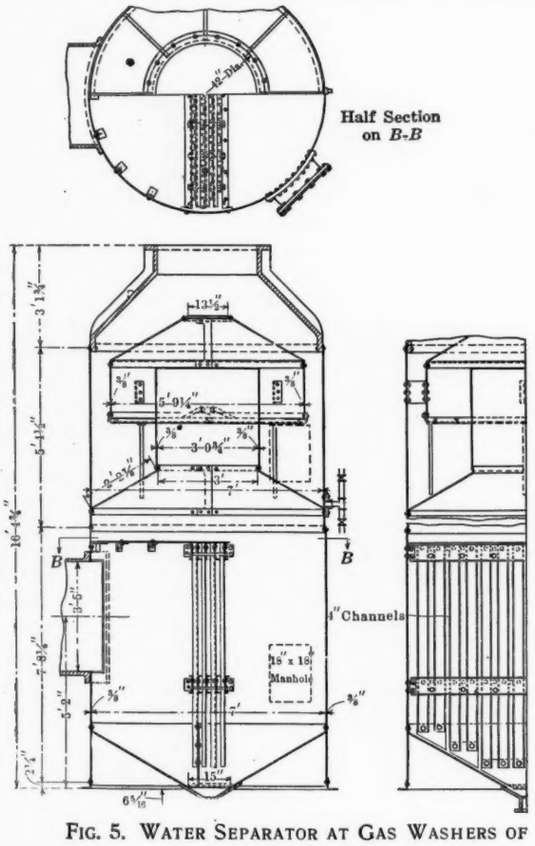


FIG. 5. WATER SEPARATOR AT GAS WASHERS OF FURNACES 3 AND 4

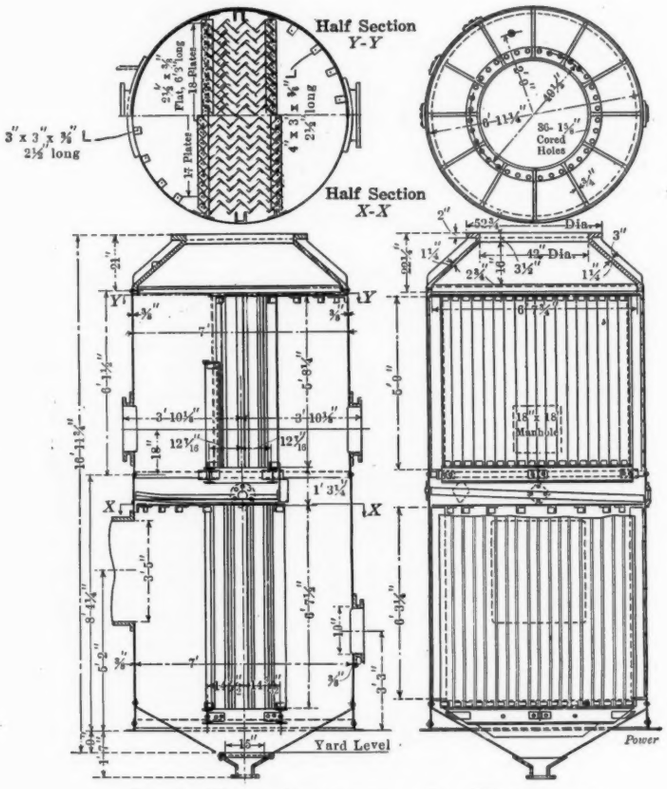


FIG. 6. WATER SEPARATOR AT GAS WASHERS OF FURNACES 5 AND 6

The cleaning of the gas at washers Nos. 1 and 2 is less complete than that at washers Nos. 3 to 6. At the two former the dust content in second-washed gas as delivered averages about 0.035 grain per cubic foot. At gas washers Nos. 3 and 4 the average dust content is about 0.328 grain per cubic foot of cooled gas, 0.061 grain in first-washed, and 0.022 grain per cubic foot in second-washed gas.

The gas supplied through the 30-inch main to power house No. 1, therefore, contains more dirt and moisture than that delivered from washers Nos. 3 to 6, in consequence of which there is more trouble with dirt at the power house than at the blowing-engine houses. Moreover, the long 8-foot gas headers at the blowing-engine houses are of probable value in taking moisture out of the gas.

Discussion

In the discussion of Mr. Coleman's paper, George D. Conlee paid tribute to the painstaking thoroughness with which the tests described in the paper had been made and brought out some interesting points regarding some indicator diagrams which were shown in an appendix to the paper.

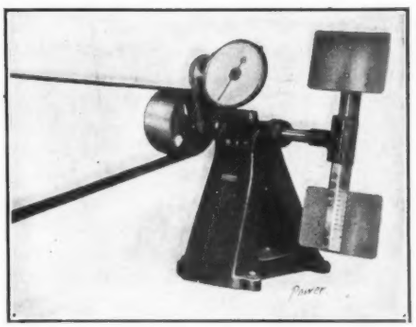
Louis Doelling, vice-president of the De La Vergne Machine Company, said that the troubles that had been experienced with these engines were due partly to errors in design but largely to dirty gas. The dust in the gas now runs from 50 to 80 milligrams per cubic meter and four years ago it was often as high as 200, whereas good practice puts the limit at 20 milligrams. In the early days, too, the gas was unstable; there were sudden rises of hydrogen due to water getting into the furnaces, and these produced premature ignitions, causing excessive wear and repairs. With clean gas a cylinder ought to run five years before requiring reboring.

The high gas consumption (18,500 B.t.u. per blowing cylinder horsepower-hour) of the Lackawanna engines, Mr. Doelling explained, was due largely to the excessive power consumed by the gas and air pumps and the loss of gas during the final half of the scavenging period. In later engines the pump loss is much smaller by reason of the use of poppet valves and the reduction of the maximum pump pressure from 10 to 4 pounds per square inch. The later engines regulate much more closely also, due to the control of the air and gas taken in by the pumps instead of bypassing the pumps; this gives sufficiently close regulation even for cotton-mill work.

George A. Orrok explained that the Lackawanna power houses are situated unfortunately in that they are between a row of blast furnaces and a row of Bessemer converters, each row being about 60 feet from the power houses. The result is that coke and ore dust from the furnaces and steel dust from the converters are constantly blown into the engine rooms and settle all over the engines. He expressed wonder that the engines would run at all under the existing conditions.

Fan Dynamometer for Testing Stationary Engines

The fan dynamometer designed by Joseph Tracy, the well known automobile expert, for testing small gasolene engines* has been adapted by Mr. Tracy to the testing of stationary gas and oil engines of moderate output. The accompanying picture shows the machine as made for this purpose. It is equipped with a pulley of a diameter which is a submultiple of the diameter of the engine pulley and the fan arm and vanes are made so that the vanes can be adjusted toward and away from the shaft by small increments, according to the amount of power to be absorbed. The arm bears a scale for each fan vane, and the dial of the indicator is graduated



Tracy Fan Dynamometer

with a power scale corresponding to each position of the vanes on their scales. Consequently, the power absorbed at any setting of the vanes can be read directly from the indicator dial.

Because of the difference between the diameters of the engine and dynamometer pulleys, the engine speed cannot be read directly from the speed scale, but by using a convenient diameter ratio, such as 3, 4 or 5, the engine speed is, of course, calculated instantly from the dial indication.

It is evident that there must be a slight loss in both speed and power by reason of the belt slippage. For this reason Mr. Tracy does not recommend the dynamometer for making close efficiency tests or fuel-economy tests. It is intended for determining approximately the maximum ability of an engine and for supplying an artificial load while "running in" an engine or testing its endurance.

*Described in Power, January 25, 1910.

Gas Power Progress during the Past Decade*

By J. R. Bibbins

Since the beginning of the present century there has been considerable progress in the field of gas power. In size, engines have developed from single-acting units of 100 horsepower or so to double-acting tandem units of 4000 to 5000 horsepower, designed for and operating under the same conditions as the steam engine. Producer units have developed from a diminutive 50-horsepower size up to 1000 horsepower or more and the problem of gasifying bituminous fuels is well along toward solution; the practice is tending in the direction of the induced-draft type for both large and small units, with the self-contained vaporizer. Opinion is somewhat divided on the tar question but all efforts are directed toward the elimination of this undesirable byproduct by gasifying it.

In its varied applications the direct-combustion principle has indeed achieved success. The gas engine has made possible the submarine, the motor boat, the automobile and the aëroplane. The aëroplane motor affords an object lesson in the results of high rotative speeds. The motor of the "Demoiselle" weighs only 3¼ pounds per horsepower and the "Gnome" aëroplane motor weighs still less.

The gas-electric motor car is another interesting development in railroad practice for service on extensions or suburban branches where steam locomotives are not warranted. The car is propelled by standard railway motors supplied from a generator driven by a compact 8-cylinder gasolene engine located in the cab. Sixty-seat coaches usually make a schedule speed of 25 miles an hour, including stops, and demonstration runs have shown a fuel economy of 0.36 to 0.48 gallon of gasolene per car-mile.

In the way of conservation, there is the utilization of blast-furnace gas, coke-oven gas and byproduct oil gas from refineries, as well as the use of low-grade fuels unsuitable for steam making. A power plant is now being built for utilizing waste cupola gas, which has a heat value of about 50 B.t.u. and requires about 300 pounds compression in the engine.

By reason of its high efficiency the gas-driven pump has been adopted for water-works service, sewage disposal, hydraulic excavation and long-distance gas transmission. The absence of stand-by losses makes it equally desirable in emergency service such as fire protection, auxiliary power supply and canal-lock operation.

*Abstract of chairman's address to the Gas Power Section, A. S. M. E., at the annual meeting, December, 1910.

Gas-engine Details

The development of the heavy-duty double-acting gas engine has been accompanied by certain interesting features. The side-crank type has been generally preferred to the foreign center-crank construction. Dry metallic-rod packing has been substituted for the elaborately water-cooled kind. Valve mechanism has been simplified by using a single cam to open both the inlet and the exhaust valves. Mixing is now done only at the inlet valves, minimizing the results of a back-fire and contributing to uniform mixture quality at all valves by eliminating fluid-inertia effects.

The electromagnetic igniter has found much favor by reason of its simplicity and the feasibility of using several igniters in each combustion chamber without entailing complex mechanism.

The series system of water circulation has reduced water consumption and also the troubles from the sweating of rods working in high-sulphur gas. The foreign practice of cambering piston rods is not followed here. With light pistons the rod flexure is not greater than is desirable to keep the sectional packing free.

Desired Features

A serious handicap in industrial work is the inability of the gas engine to supply enough exhaust heat to warm a factory. Some progress has been made with the exhaust heater but the 5000 or 6000 B.t.u. per brake horsepower-hour available from an engine is not sufficient to do the work. Some system including an auxiliary gas-burning heater must be worked out.

More convenient and practical methods of measuring the volume and heat value of gases should be provided. Some large plants have adopted the venturi meter, but even this simple apparatus is sensitive to deposits in the throat. A continuously recording calorimeter is greatly needed, and some progress is being made in this direction.

There is a disposition to discount the demand for large engine and producer units. With steam-turbine units increasing rapidly in size the gas-power industry must respond in kind or have the gas engine remain an auxiliary for special conditions.

Education of the operator, the salesman and the manufacturer is essential. The great mistake is made in partial education—an incomplete understanding of the conditions, a make-shift equipment and a jealous guarding of knowledge of defects. The results are loss of confidence, dissatisfaction and failure.

Power from Crude Oil

Development of the oil engine has made great progress abroad since the expiration of the basic Diesel patents. Two of the principal builders have turned out 250,000 horsepower in engines, some of which rated as high as 1000 horsepower per cylinder. The smaller engines mostly work on the four-stroke cycle, but above 1000 horsepower the two-stroke cycle prevails.

In the various experiments with oil-gas producers the small progress has been discouraging. Two systems have been used, the retort and the partial combustion. In the former, difficulties with carbon deposition in the retorts are encountered; in the latter, excessive production of lamp black. Both are hopelessly low in efficiency as compared with the oil-burning steam plant. A large oil-gas plant in California, operating gas engines as water-power auxiliaries, endeavors to apply to power purposes mixed gas, consisting of part retort and part carbureted water gas, utilizing the carbon deposits of the former as briquets in the latter process. In this mixed gas, the hydrogen content is kept down to about 30 per cent., but in the oil gas it is very much higher, 40 to 60 per cent. For straight power purposes the combustion producer seems more promising both in simplicity and efficiency.

Peat

We have looked to Canada for important developments in the use of peat, but private experiments have failed so signally that the Government has started a peat-manufacturing and power plant to demonstrate the process on a commercial scale and reëstablish confidence in this industry. Director Haanel, of the Bureau of Mines, thus summarizes his investigations: Artifical drying processes have failed commercially and a machine process must be substituted for the manual labor. The department is, therefore, proceeding along European lines of established success. He states that Russia alone produced 4,000,000 tons of peat fuel in one year—1900. Peat containing not over 25 to 30 per cent. of water has been found an ideal fuel for gas-producer work, requiring no additional steam and being quite free from high temperature and clinker. The long series of fuel tests at Montreal have served to confirm the results of our Government tests on lignites in demonstrating the great possibilities of these lignite deposits, especially in the Canadian northwest.

Gas Engine Troubles

A remarkable array of facts on gas-engine troubles was presented by Charles Kratsch in a paper before the National Gas and Gasoline Engine Trades Association during its recent meeting at Racine. The information was collected from the trouble calls arising from one hundred engines ranging from one horsepower to 125 horsepower multiple-cylinder verticals for generating electric current; the makes included nearly all types from the old slide-valve Otto of thirty years ago up to the modern types which are on the market today.

Seven per cent. of the failures came under the classification of causes due to installation. Among these causes were "engines installed by the purchaser to save first cost; gas bag too far from the engine; no coil in the ignition circuit; cooling water reduced so that the engine overheated, and cam-shaft gears not in mesh properly."

Thirteen per cent. of the failures were classed as causes due to fuel. The principal one of these was the location of the supply tank too far from the engine to feed sufficient fuel at all times; faulty fuel supply due to carbureters or mixing valves; fuel-supply pumps, or clogged piping was another.

Ten per cent. of the failures were due to lack of proper instructions for operation, some of the results of which were too much or too little gas; no cylinder oil; too much or too little cooling water; weak or dead batteries; defective or improperly adjusted vibrator on spark coil; parts put together wrong after the Saturday night tinkering.

Five per cent. were classed as due to faulty construction, under which head came defective parts blamable to design, such as crank shafts of too small dimensions; insufficient valve area; valves opening late or for too short a time; not enough lift to valves for perfect mixture or clear exhaust; bad gasket faces, causing water leaks.

Seventeen per cent. came under the head of causes due to natural wear and accident, of which the following were cited: worn cylinders; shafts cut, sprung or crystallized; valves needing regrinding; governor fingers worn out; lost motion in bearing brasses; loose flywheel; gaskets blown out; crystallization of connecting-rod studs; general overhauling; engine totally wrecked.

Nineteen per cent. were classified as causes due to ignition troubles, as follows: parts inside the engine damaged by wear and neglect; movable electrodes worn out; igniter plugs requiring new bushings, new points, springs, etc.

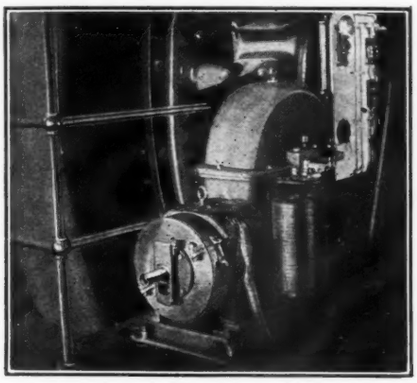
Twenty-nine per cent. of the failures were due to equipment and accessories and nearly one-half of these were troubles that could have been anticipated and shutdowns eliminated if an extra igniter had been furnished. Eleven of the 29 per cent. in the "accessories" class were ignition troubles caused by poor wire, defective switches, bad installation of wiring, poorly connected terminals, wires short-circuited through poor insulation, burned-out coils, poor magnetos and cheap batteries. The remainder of this division were troubles due to igniter points being burned off by excessive ignition current, the current being supplied by small generators driven at too high speeds or from lighting circuits presumably of too high voltage.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment

Electrical Barring Machine

The accompanying engraving shows a simple and compact motor-driven appliance devised by the American Ship



ELECTRIC BARRING MACHINE

Windlass Company, Providence, R. I. to do away with the difficulty attending the turning over of a large engine at the works of the Stanley Company, Bridgewater, Mass. The illustration shows the barring machine geared to the flywheel of a 32- and 36-inch by 60-inch engine, nominally rated at 2000 horsepower, and running at 75 revolutions per minute.

The machine consists merely of an electric motor worm-geared to the "barring" shaft, which carries on its outboard end a spur pinion meshing with an internal gear bolted to the inner rim of the 20-foot 75-ton flywheel. When the machine is not in use, the pinion is drawn out of mehs with the gear by means of a hand lever which slides the complete machine along rails on the bedplate. The motor is an 11-horsepower Westinghouse machine, which runs at 700 revolutions per minute on a 220-volt direct-current circuit. It turns the flywheel through one revolution in about a minute. It is especially wound for heavy starting torque and is provided with a reversing controller having five forward and five reverse positions.

An alarm bell is so connected with the outfit that it rings during the entire time that the pinion is in contact with the gear on the rim of the flywheel, thereby reminding the operator to throw the pinion and gear out of mesh before starting the engine.

The Electrical Equipment of a Large Department Store

BY NORMAN G. MEADE

The Gimbel building, located at the intersection of Broadway and Sixth avenue between Thirty-second and Thirty-third streets, New York, is the largest one in the country devoted to retail mercantile purposes. The electrical installation consists of approximately 25,000 incandescent lamps, 36 electric passenger elevators, 10 electric freight elevators, and package conveyers, ventilating fans, pumps and the multitude of an auxiliary apparatus that is usually included in the equipment of a large building. The supply of electricity is obtained from the Waterside plant of the New York Edison Company.

Owing to the size and nature of the installation it was deemed expedient to install a regular substation in the building to receive current directly from the Waterside stations, which are about two miles distant. Consequently, three independent feeders were laid from the sta-

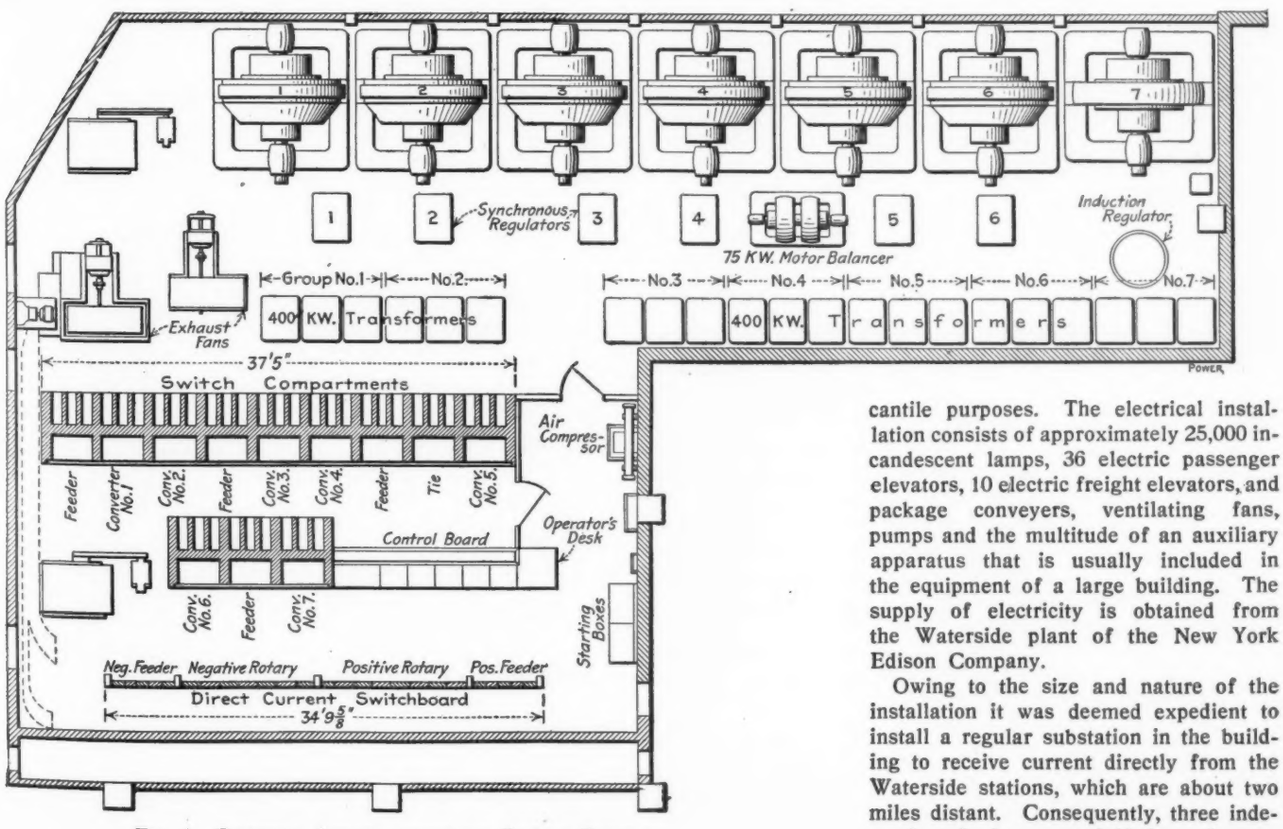


FIG. 1. GENERAL ARRANGEMENT OF GIMBEL PLANT

tions to the Gimbel building, carrying three-phase currents at 6600 volts and 25 cycles frequency. The equipment of the substation consists of six 1000-kilowatt Westinghouse six-phase rotary converters of the synchronous-regulator type, and one 1000-kilowatt General Electric six-phase rotary converter, controlled by an induction regulator. All of the converters deliver direct current at 250 volts. The high-tension alternating current is stepped down by twenty-one 400-kilowatt air-blast transformers, three to each converter, connected in delta at the primary terminals, and double delta at the secondary terminals to obtain six phases. In rotary converters of large capacity higher efficiency and more economical distribution of copper are obtained with the six-phase winding than with the three-phase winding.

The transformers are set in a single row across the room, over a large conduit or air duct. The air blast is furnished by two motor-driven blowers, one of which is held in reserve. The heated air leaving the transformer dampers is removed from the room by a motor-driven exhaust fan and discharged outside the building.

The high-tension switches are inclosed in masonry and are all of the remote control type, operated from the control switch board, as indicated in Fig. 1. The feeder and rotary converter switches are motor operated and the selector switches are operated by solenoids.

The direct-current terminals of the rotary converters are connected to a direct-current switchboard provided with separate busbars for the rotary converters and the feeders. This board is connected to the house board controlling the building load and is also provided with a connection to the low-tension direct-current street mains of the Edison system.

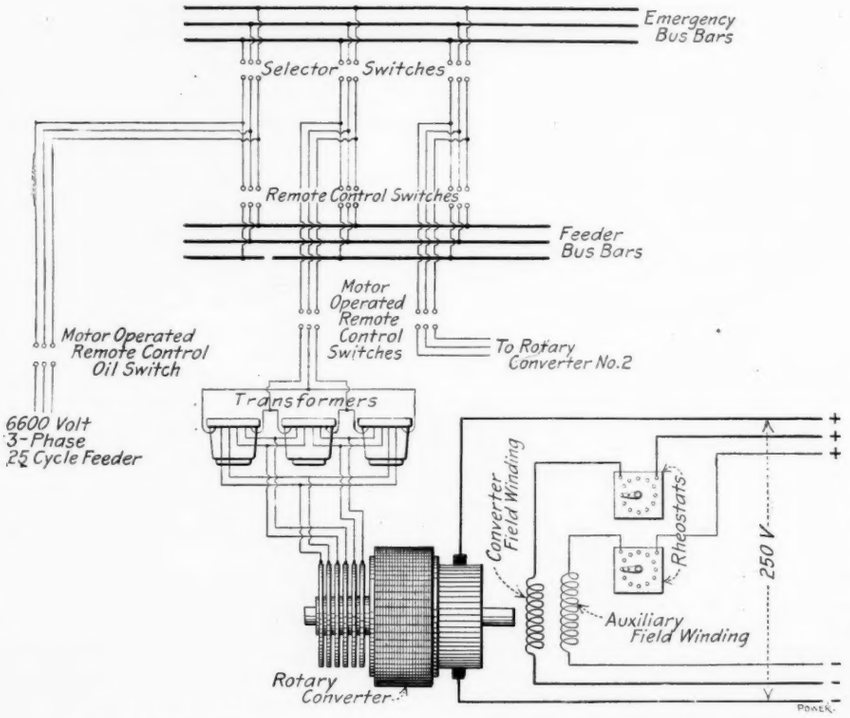


FIG. 2. DIAGRAM OF HIGH-TENSION ALTERNATING-CURRENT WIRING

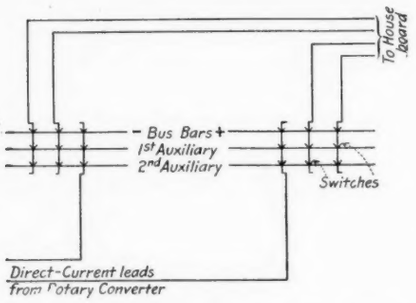


FIG. 3. SCHEMATIC DIAGRAM OF DIRECT-CURRENT BUSBAR CONNECTIONS

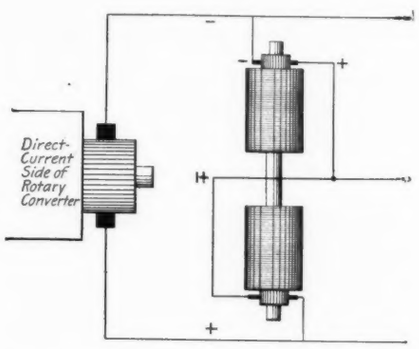


FIG. 4. CONNECTIONS OF MOTOR BALANCER

Fig. 2 shows the elementary connections of the high-tension alternating-current wiring. The alternating-current switchboard is provided with two sets of busbars; the feeder busbars are divided into four sections but the emergency busbars extend the whole length of the board. Each feeder normally supplies two rotary converters. The seventh converter may be used in place of any of the other six, or may be used to assist other Edison substations by means of a tie-in feeder.

Each main feeder is connected to the alternating-current switchboard by a remote-control motor-operated high-tension oil switch, and in turn may be connected to the emergency or the feeder busbars by means of remote-control selector switches. Each converter is connected by means of a motor-operated remote-control switch and may be supplied with current from the emergency or the feeder busbars by closing the proper selector

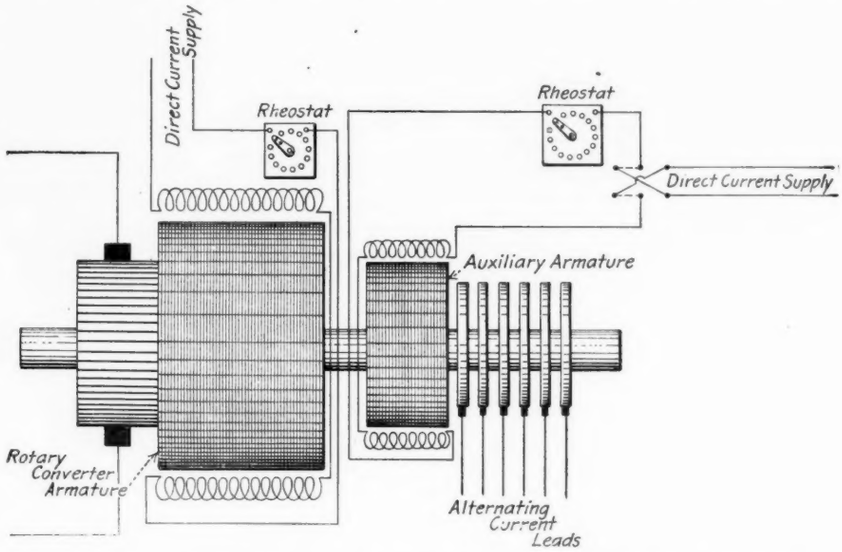


FIG. 5. DIAGRAM OF SYNCHRONOUS REGULATOR TYPE OF ROTARY CONVERTER

switch. This arrangement of connections is very flexible and makes it possible that in case of emergency one rotary converter can be substituted for another almost instantly. All of the converters are started from the direct-current side in a manner similar to the starting of a direct-current motor and when the synchronizing lamps indicate synchronism the high-tension converter switch and the proper selector switch are closed.

A schematic diagram of the main direct-current switchboard is shown in Fig. 3. It is provided with a main positive and two auxiliary positive busbars and one main and two auxiliary negative busbars. As will be seen from the diagram, which shows the direct-current leads from one over-all voltage. Fig. 4 is an elementary diagram of the arrangement. The commutator of only one converter is shown, but the others are all connected in parallel with the one shown and do not affect the operation of the balancing set. The set is capable of taking care of 150 kilowatts of unbalanced load.

One of the most interesting features of this substation is the synchronous-regulator type of rotary converter, which differs considerably from the ordinary type of converter. The synchronous-regulator type of machine is provided, in addition to the usual component parts, with an alternating-current generator built integral with it and having the same number of poles as the main field magnet

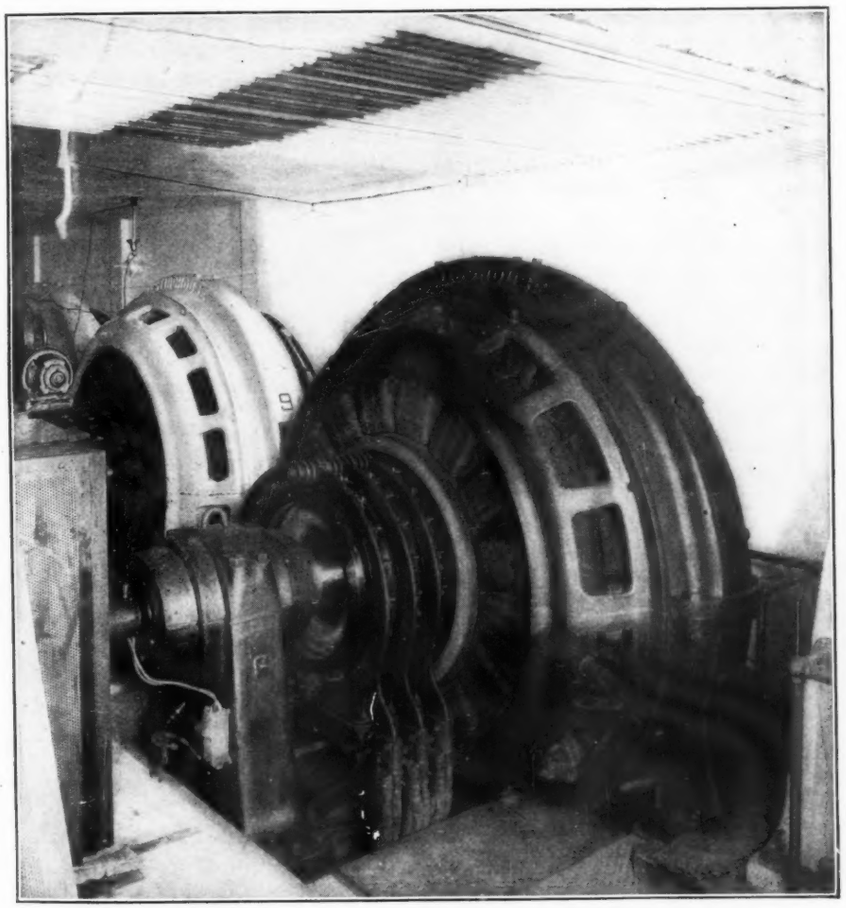


FIG. 6. ONE OF THE SYNCHRONOUS-REGULATOR CONVERTERS

converter only, any one of the three busbars on either the positive or the negative side may be used by throwing in the proper switches. Likewise the feeders to the house board or to the street network may be connected to the various busbars at the option of the attendant.

The building is wired throughout on the three-wire plan, and as the converters deliver the overall voltage (250) a motor balancer set is provided. This consists of two direct-current machines coupled mechanically and connected up just as they would be in a straight direct-current plant where the generators give the

of the rotary converter. The armature of the alternator is mounted on the converter shaft and its winding is connected between the main armature winding and the collector rings. When the polarity of the field winding of the auxiliary alternator corresponds to that of the main converter field winding the voltage of the alternator armature is added to that of the rotary converter. Consequently, the ratio of alternating to direct voltages at the main terminals is increased and the direct-current voltage is lowered. If the polarity of the auxiliary field winding is reversed the electromotive force of the alternator or auxiliary armature opposes that of the rotary converter armature and the resultant alternating electromotive force is reduced correspondingly, with reference to the direct-current voltage; the latter is therefore increased, because the alternating voltage is fixed by the supply circuit and changing the converter ratio can change only its direct-current voltage. The degree to which the direct-current voltage is raised or lowered by the auxiliary armature is controlled by a simple rheostat in the auxiliary field circuit. This rheostat is operated by a motor which is controlled from the switchboard. The polarity of the auxiliary field magnet is changed by a reversing switch (shown diagrammatically in Fig. 5) also operated by remote-control from the switchboard.

Electrical Accidents Due to Carelessness

BY HOWARD S. KNOWLTON

No course of instruction in the university of hard knocks conveys a more lasting lesson than that of acidents and misfortunes. Year by year the study of casualties and their prevention leads to safer operating conditions in every branch of industry. Repeated reference to these matters is justified on the ground of economy in operation no less than that of preventing human suffering. New conditions are constantly arising and new men entering the field. The older men engaged in the operation and construction of engineering systems tend to overlook the sources of danger, while the fresh recruits frequently fail to realize the disasters which can result from small beginnings. This is peculiarly the case in the electrical industry. A review of the causes of a number of electrical fires and accidents which occurred within one year in a single municipality may well emphasize the importance of trouble prevention over a wider field. In the following paragraphs the essential features are given of troubles occurring almost entirely as a result of negligence, which in most cases were handled without serious financial loss, but which might easily occur anywhere and produce less fortunate results.

SOME ELECTRICAL FIRES

The careless leaving of an elevator motor connected to the supply main when the machine was stopped for the night roasted out the field winding and started an incipent fire, which fortunately was discovered before anything but the motor was injured. In another case, for lack of proper attention the brushes on a 1-horsepower motor became so badly worn that violent sparking occurred at

the commutator and oil, dirt and sweepings, which had been allowed to collect, became ignited. In a third instance, a defective rheostat was in service in a basement, and when the operator of the plant started the motor-driven machinery serious overloading of the motor occurred; the protective equipment did not operate quickly enough to prevent a burnout, and the machine was badly injured. While the fire in none of these cases spread to cause any serious damage in the vicinity of the electrical equipment, it was due only to the prompt discovery of the situation through the smell of burning insulation and rapid accumulation of smoke that the fire loss was small.

Careless handling of wires during the installating of electrical apparatus is responsible for much trouble. A slight fire in a basement hallway was caused by linemen working outside on overhead wires, who permitted the circuits to sag sufficiently to make contact with a trolley wire and street lamp post at the same time, permitting current to pass from the wire to the post and thence by means of a gas main to the adjacent building, where a water pipe was in contact with a gas pipe running to a gas bracket on a side wall. Holes were melted in the gas pipe, the gas became ignited, and a small fire occurred. This occurrence illustrates the ease with which the improper handling of electric work may lead to troubles at some distance from the immediate locality where the negligence happens. In the same class of accidents are those where derrick guy wires and chains are unintentionally brought in contact with either the feed or trolley wires of an ordinary direct-current railway service, one side of which is grounded. The failure to screw the plugs of fuse cutouts tightly home is another efficient cause of trouble. The poor contact causes heating, which becomes intense through arcing or lowered conductivity, and if the fuse does not blow enough energy may be released in a small area to produce a disastrous blaze. One fire last year was caused by a porter leaving a coat and pair of overalls hanging over a cutout of the cartridge type. As a result of loose connections at the fuse clips, probably caused when the clothing was hung up, arcing and heating of the contact resulted, and the garments were set afire.

A small fire in a garage started on a table where several 6-volt ignition batteries were being charged. The batteries were left alone during the night and some had boiled over. Current leakage occurred between two of the jars, the liquid on the surface of the table having acted as a conductor. Here again the presence of heavy smoke warned the occupants of the building that something was amiss.

Two other representative fires were due to unrelated causes. One occurred through the burning out of the armature of a ¼-horsepower motor, which was due to the bearings becoming dry through inattention; the second was an out-of-door flare-up produced by the end of a wet rope coming in contact with one side of a series arc-lighting circuit. The rope had been used by the lineman of a signaling company to temporarily fasten a new cable in position, and the current was grounded on a rainy night.

Personal Injuries

In spite of the frequency with which workmen in power plants and on the structures of elevated railways are warned against making short circuits, severe personal acidents of this kind occur repeatedly each year, and almost always through carelessness in the use of tools. Where one side of the circuit is grounded the trouble generally reaches a more acute stage in point of arcing than where the circuit is metallic throughout. Among the accidents of this kind which occurred last year in the community in mind was one where two men, both regular employees of the company, received severe burns about the face and arms while at work in a power plant of the 600-volt railway type. Their injuries were due to a short-circuit through a wrench which they were using in the removal of an iron pipe coming in contact with the live metal of a fuse board at the time when one end touched the pipe.

In another case a lineman was soldering a very heavy cable used for railway service, when the metal ladle which he was using came in simultaneous contact with the joint of the live conductor upon which he was working and a grounded pipe carrying compressed air.

In a third case the workman accidentally brought a wrench in contact with a live third rail while engaged in loosening nuts on one of the running rails of the track. Heavy burns about the face, hands and arms resulted. Two other accidents arose from the careless use of tools in the vicinity of a third rail and feeder installation. The first was caused by a carpenter's saw engaged in cutting off the end of a tie coming in contact with the third rail and grounded elevated structure; the second by a hammer which was being used in a cable box coming into simultaneous contact with a bare live connection and a bolt which was in contact with the structure.

Bad facial burns were received by two wiremen as a result of a short-circuit due to their own carelessness. They were to remove an unused and dead wire from a conduit, it being necessary to cut the wire before its withdrawal; by mistake they attempted to cut the wrong wire, which was alive and a short-circuit was caused by the cutters making simultaneous contact with the live wire and the pipe. A similar accident occurred in a power station where a workman was inserting a copper filler between the plates of a busbar structure, simultaneous contact being made between the busbar and the grounded framework supporting it.

Low potential systems are capable of causing personal accidents no less than high voltage installations. Severe burns occurred on the hands of an experienced installer as a result of a short-circuit caused by a monkey wrench on the shunt connections of a low potential motor-generator set. The use of a jackknife in making temporary connections at a junction box also led to a short-circuit, which caused painful burns. Care is equally necessary to avoid trouble in the installation of insulating materials in the neighborhood of busbars. In one recent instance severe burns resulted from a heavy short-circuit which occurred when a workman was attempting to put a bolt through a piece of alberene stone and angle iron for the purpose of fastening the stone to the angle iron. The bolt end came in contact with a live storage-battery busbar, and the current grounded through the bolt and angle-iron hanger.

LETTERS

Identifying Alternating and Direct Current

Referring to H. Priestley's inquiry in the December 6 number for a method of finding out whether the current in a lamp socket is alternating or direct, I would suggest that this can be ascertained by holding one pole of a permanent magnet against one side of the globe of an incandescent lamp while the lamp is lighted. If the lamp is supplied with direct current the magnet will attract the filament to one side. If it is alternating current the lamp filament will vibrate, due to the alternations.

R. L. Mossman.

Tampa, Fla.

[Exactly the same suggestion has been received from E. F. Potter, Urbana, Ill., and Roy Stolp, of Chicago.—Editor.]

I believe that the liquid method is as simple and reliable as any. Take a glass of water and put a pinch of salt in it. Insert the two ends of the wires in the glass, which should be in series with the lamp on the circuit to be tested. With direct current the negative wire will give gas off freely in the form of bubbles, while with alternating current both wires will give off gas to some extent, but in equal amounts.

Louis J. Gorilla.

Ironwood, Mich.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words wanted

New Engine Required Lining Up

A new 12x16-inch engine was installed in a sawmill and, naturally, the management expected things would run smoothly. But for some time that engine did some queer stunts. It ran under, and the under guide ran hot. The engine seemed to labor hard and did not develop its rated power.

I was called upon to see what could be done and I found that not only the bottom guide of the engine ran hot but that the engine heated in other places also, although there seemed to be enough loose play. The crank box could be shaken at some parts of the stroke, but at other parts would be tight.

I put a line through the cylinder and got a surprise. The engine had a self-contained base, the outboard bearing and frame in one piece, and I could hardly think that the shaft would be out of line on this new engine, but in getting my line true with the cylinder base, I was surprised to find that the line was up above the shaft center, and that the cylinder was out of line with the guides.

Then, ignoring the cylinder, I tried to line up the shaft with the guides, and found that they were larger at the cylinder end than at the crank end, and were also smaller at the center than at either end.

There was no boring bar within miles of the plant, but, on looking around, I ran across a shaft that was of the same diameter as the stuffing box of the piston rod.

A box was found to fit this shaft and it was clamped to the crank disk. The shaft was then run through the stuffing box and the box on the disk.

I next made some clamps of heavy, flat iron to hold the tool, and then threaded a bolt long enough to feed the bar through the guides. A heavy bar of iron was placed across the cylinder end and was secured by two of the stud bolts. This end bar had a threaded hole in the center for the feed screw.

A handle was clamped on the crank end of the boring bar and two men were set at work turning the bar. With a set collar on the bar to feed against, I bored out the guide, and made a pretty good job of it, considering the tools I had to work with.

Next, the cylinder was removed from the frame in order to find out what made it hang down out of line, and I found that the frame had been faced off out of true. I did not care to try to face it up with the boring bar I had improvised, nor did I like the idea of putting any kind of soft packing in such a place. Then I thought I would try a plastic cement, and some was put in the opening left between the frame and the cylinder, and after lining it up I had a first-class job.

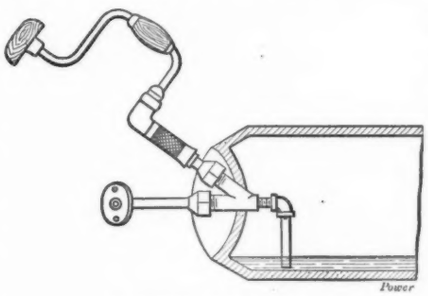
It was necessary to rebabbitt the main bearing to get the main shaft in line, but after this was done the engine ran like a new machine should.

JAMES W. LITTLE.

Fruitland, Wash.

Bit Brace as a Wrench

The following method of using a common bit-brace, tightened firmly on the valve stem of an ammonia drum, as shown in the illustration, may be of some



OPENING THE VALVE

value. I have tried different styles of wrenches, but for quickness and sure control in opening the valve I have found nothing to equal the bit-brace.

R. F. PROPPITT.

Chicago, Ill.

Does It Pay to Waste Heat—Sometimes?

The writer had an interesting discussion a while ago with a prominent power-plant engineer in regard to a rather unique installation which has recently been completed for a large New England textile mill. Briefly, the plant consisted of three main turbine-driven units of 500 kilowatts each, with separate condensing units; four cooling towers with individual motor-driven fans; and the usual complement of auxiliaries of boiler-feed pumps, circulating and service pumps and a pump for the heating system.

Originally it was intended to utilize the condensed steam from the main units for heating the mill, there being no other use for which it could be utilized except heating the boiler-feed water. When this plan was presented to the owner he claimed that the "time factor" as regards heating had not been considered—that is, that the time he wanted the heating done was between 2 and 9 or 10 am., the greater part of which time the mill would not be running.

As a result it was decided not to utilize the exhaust steam for heating. Instead, jet condensers were installed for the main units, the boiler-feed suction being taken from the return line to the cooling towers. This is passed through a feed-water heater and was raised from about 90 degrees to 200 or 210 degrees by the exhausts from the various auxiliaries. The turbines operate at about 28 inches vacuum.

The mill heating system is of the hot-water, forced-circulation type, the water being heated by live steam except for the excess from the auxiliary exhausts which is utilized when the plant is running, by passing it through a tubular heater through which the water for the heating system is pumped before going to the live-steam heater.

At first glance this layout may seem uneconomical in its utilization of heat units, but upon careful inspection it appears that, all factors being considered, the total cost of producing power may be less than in a plant where the effort is made to utilize every possible B.t.u. from the coal. There is a distinct economy in maintenance cost effected by the use of jet condensers and, furthermore, less circulating water is required. In this particular instance the water goes to the feed-water heater at about 90 degrees, whereas if it were taken from the condensed steam in a surface condenser it would reach the heater at perhaps 100 degrees with the turbines exhausting at a little less than 28 inches vacuum.

It seems very reasonable to expect that the cost of saving the heat represented by this 10 degrees might possibly be greater than the immediate loss due to throwing it away. As the plant in ques-

tion has only been operating a comparatively short time, accurate figures are not yet available. The point brought out here may, however, be food for thought for both designing and operating engineers.

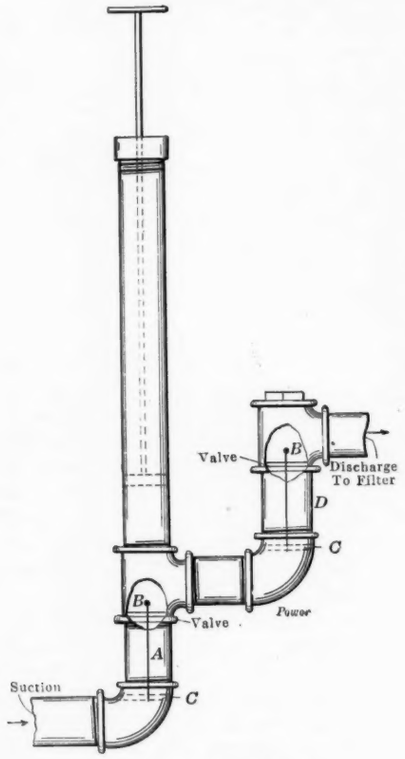
H. M. WILCOX.

Boston, Mass.

A Handy Oil Pump

When I took charge of a shift in a certain plant, I noticed that a compressor was fitted with a neat looking homemade waste-oil pump.

This compressor ran day and night



DETAILS OF OIL PUMP

and, as it was of the vertical type, the waste oil drained into a receptacle in the base of the machine. Unfortunately the builders had neglected to provide means to remove the waste oil. Therefore, a hole was drilled through the side of the compressor wall and a pipe led through to the oil trays inside the base. The pump was then permanently attached to the machine, and the discharge pipe led to the oil filter. The cylinder of the pump was made of polished brass pipe and the other 1-inch fittings were given a coat of black japan.

The tops of the nipples *A* and *D* (see illustration), were filed flat and made excellent seats for the valves. Ordinary hard-rubber bibs were used as valves. A long nail extended through each valve and nipple and was riveted to a spider at *C*. This allowed the valve the proper amount of lift.

WILLIAM WATT.

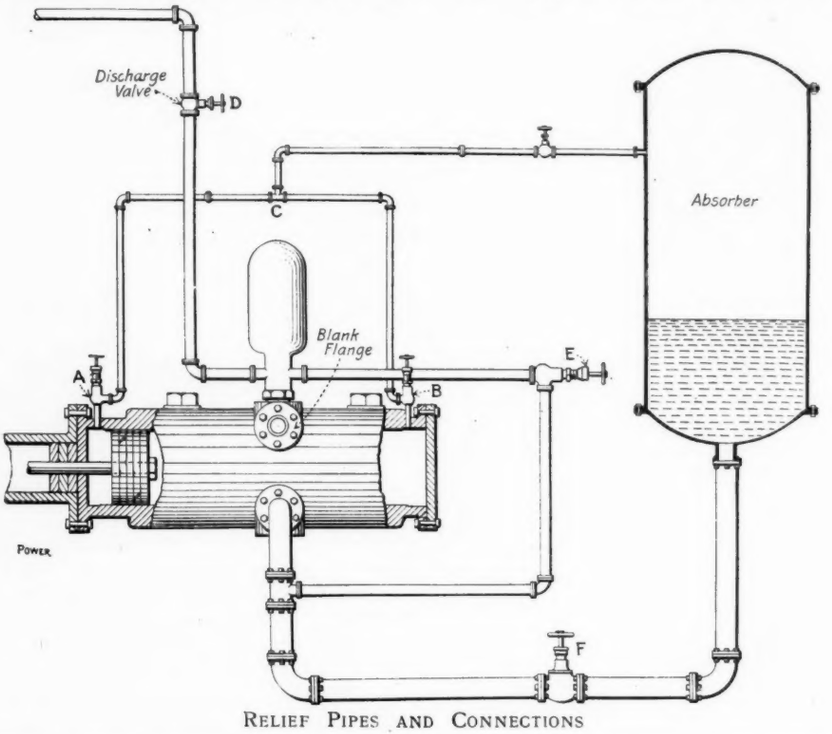
Lambton Mills, Canada.

Relieving an Aqua Ammonia Pump

Aqua-ammonia pumps are heir to the common ailment of becoming gas-bound, and when this happens, the engineer has no positive means of relieving the gas. Putting cold water on the pump and forcing the absorber pressure up does no good, as the pump will repeatedly become gas-bound when the pressure returns to normal. Sometimes it will take hours to get everything regulated so that the pump will work steadily, and during this time the temperature is rising.

The accompanying sketch shows a scheme that I applied to a pump with success. I drilled and tapped each end of the pump at the counterbore for a ½-inch connection and screwed in the valves *A B*. From these I ran two lines to the tee *C* and a line from *C* to the absorber.

When the pump plunger moves to the end of the cylinder, as shown, it compresses the gas and by opening the valve *A* a large portion of the gas will be driven into the absorber. When the plunger moves to the other end of the cylinder there is so little gas on the side of the plunger just relieved that the pressure will fall rapidly until it is below the pressure in the absorber. Then the liquor will be forced in from the absorber and the pump is immediately relieved without interfering with the equilibrium of operation and without a loss of temperature.



RELIEF PIPES AND CONNECTIONS

The bypass from the discharge to the suction is very convenient when packing the pump. By closing the valve *D* and opening the valve *E*, after drawing a vacuum on the absorber, all of the ammonia is removed from the pump. Packing can then be carried on without being obliged to run from the ammonia fumes, but it will be necessary to close the valves *E* and *F* before starting to pack.

J. J. NASH.

New Haven, Conn.

Why Did the Feed Pipes Clog?

In a power plant in Nevada where I was employed for eighteen months prior to dismantling the plant about a year ago, we had seven water-tube boilers of various sizes.

The water was very good and the condensed steam from the jet condenser passed over a cooling tower and back to the boilers.

A 4-inch header extended across the boilers and a 2-inch pipe ran from the header to each boiler; there were five turns in the 2-inch pipe and the entire piping was of wrought iron with some cast-iron fitting.

There was no scale in the 4-inch header to speak of and very little in the boilers; surface well water was used at a temperature of from 160 to 190 degrees Fahrenheit.

These boilers were kept practically free from scale; in a few months a 2-inch feed pipe would fill up with scale so that a ½-inch rod could not go through the pipe.

Will some reader explain why the feed pipes would clog up and yet no scale form in the boiler? A compound was used in the feed water.

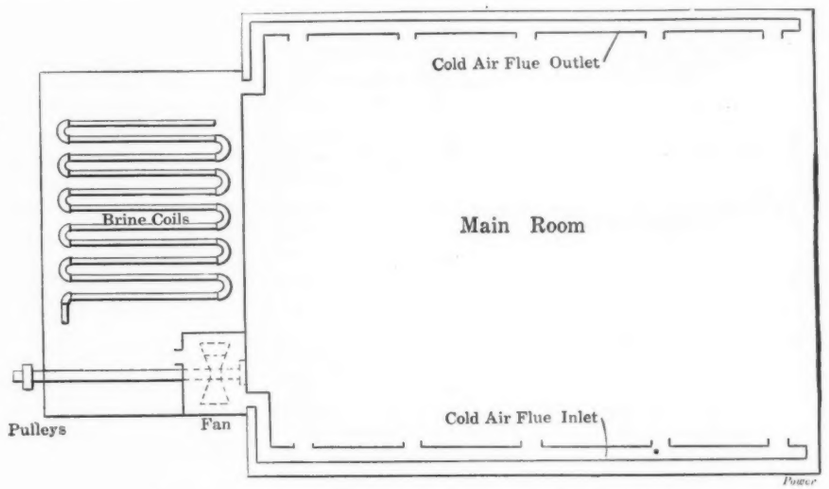
WILLIAM E. PIPER.

Farmington, Utah.

Cooling Cold Storage Rooms

In laying out cold-storage rooms the engineer has several methods of cooling from which to select.

The best method for any particular case depends entirely on the kind of work. If it is necessary to eliminate moisture from the room, the plant should be laid out with a small room connecting with the main room. This room should be practically filled with brine or ammonia coils, with the exception of a small space at each end which should be utilized for the cold-air flue. The coils should reach the entire width of the room, and extend from the floor to the ceiling, thus making it necessary for the fan to draw the air from the main room through the coils.



ARRANGEMENT OF COLD-STORAGE ROOM AIR DUCTS

A fan should be placed at one end of the room, and from this the cold-air flue should start. This flue should be made of galvanized iron, and fitted with small slides or openings about every ten feet.

The return flue should start at the other side of the small room and run on the opposite side of the main room. The floor plan is shown in the illustration.

This system will enable one to keep the main room free from moisture, but it requires more refrigerating capacity owing to the indirect manner of cooling the room. It is also necessary to clean the frost from the coils occasionally, as in time it becomes so thick as to completely shut off the circulation of air. The quickest way to rid the pipes of snow is to close the air slides to the main room, and use a hose and hot water, afterward sweeping as much of the water out of the room as possible and absorbing the rest with sawdust. The brine or ammonia is shut off during this operation.

If a little moisture will do no damage to the contents of the room it can be cooled in a better and more direct way by the brine or ammonia coils being placed directly in the room. Of these two latter methods the ammonia coils are the most direct, as the ammonia is doing no work until it has passed through the expansion valve; therefore, the heat is absorbed directly in the room to be cooled; in the brine system the heat is absorbed in the cooler, thus cooling the brine, which is then pumped through the coils in the cold room.

Comparing these two methods, the direct expansion is far superior to the brine system for four reasons:

First. The expansion of the ammonia must take place either in the cooler or in the room direct. If the expansion takes place in the room, it is bringing the temperature of the room down; if in the cooler, it is bringing the temperature of the brine down, and as the brine must be brought to a low temperature before it will do any work on the room, it is not difficult to see that the room can be cooled much quicker with direct expansion, thus making the operating expense less.

Second. A brine pump is required with the brine system, which also adds to the operating expense.

Third. The brine loses considerable in temperature in traveling to and from the cold room.

Fourth. A larger coil is required for brine, in order to obtain the same results as from the ammonia.

The brine system has one very good point in its favor, in that the ice machine can be shut down for repairs between four and five hours, and if the brine pump is kept running the room can be held at the usual temperature.

Another very good system is a combination of the brine and direct expansion. A small pipe runs through a larger one, and brine is pumped through the larger pipe, while the gas is expanded through the smaller one from the opposite end of the coil. With this system it is not necessary to have a cooler or ammonia coil in the brine tank, which is used only to hold the surplus brine.

In some plants economy requires the installation of both brine and ammonia coils. In operation the room is brought to a low temperature by the direct expansion; then the machine is shut down, and the room held at this temperature for a considerable time by circulating the brine through the room.

WALTER C. EDGE.

Philadelphia, Penn.

Ethics of the Engine Room

There is such a thing as ethics in the engine room, and to this any live engineer will agree. Ethics is defined as the science of human duty.

The spirit with which any man enters into his work largely determines his success or failure. The man who can put his whole soul into his work, and who believes his work is worthy of his best efforts, is pretty sure to succeed, even though he be a plodder. No man ever won success by working only when he felt like it.

The man whose habits of living are correct and conducive to the conservation of his physical and mental energy is able to overcome any obstacles that may arise.

No man rises above his ideals. Many a man makes the mistake of not keeping an eye on the job higher up and working and studying to fit himself for it. Then when the time comes to move up the line, he is ready.

The fire engine and the harness are ready, the men and horses are drilled and trained for action, ready to go when the alarm sounds. If this preparation were not made, what would be the result? Some day a man will be required for a job and if the man who should be promoted is not competent to fill the place, someone else will get it and the fellow who was "weighed in the balance and found wanting" will work away at his old job in a disappointed and half-hearted manner that spells doom to him, so far as advancement is concerned, unless he takes "a brace," gets the vision and pursues it with a determination to win.

A farmer fills his wagon box with potatoes and starts to market, and as the wagon jostles along over the country road the largest potatoes work toward the top. So it is with the man who really desires something better and is willing to fit himself for advancement. It is this desire coupled with determination that makes him a "large potato," capable of rising toward the top as he bumps along over the rugged pathway of life.

It is not what one might do, however, if he had so-and-so's ability and education; but it is the use that is made of the brains and energy we have that marks for success or failure. Strive for advancement; for, by the very effort, one

will have bettered his condition, even though his aims are not fully realized.

Many complaints are heard with reference to certain men having a "pull," as though they needed only to have a "friend" or in some way to "stand in" with the "boss." This is a matter that usually adjusts itself; for, though a "pull" may get a job, it will not help to "make good."

A wise man is continually learning. He looks after his employer's interests with the same painstaking care that he would expect should he be employing men in a business of his own. Some may say, "The boss does not appreciate my efforts and I am not going to exert myself to look after his interest." If the "boss" does not appreciate your efforts, someone else will. The man who fools the "boss" fools himself more. It is better for a man to outgrow his job than to let the job do all the growing.

There are certain relations that should exist between employer and employee. Not only should the employee come to his work with the right spirit, but the employer should greet his employee as a man and a coworker, show him that his efforts are appreciated and give him enough insight into the business so that he can see his own relation to the business and to his fellow employees. In doing this a man's efficiency will be greatly increased because he realizes the importance of his own particular part in the business and is filled with a desire to make the best possible showing. The employer who does this and who gives his employees fair treatment in all matters is building up a business that will be both pleasant and profitable for all concerned.

What is true in a general way in the industrial world is also true in the power plant and the engine room. This same spirit, if maintained, will make the tasks easier to perform.

The man who does not develop manhood along with his work is missing the best part of life. The practice of keeping the engine room neat, clean and orderly; the practice of economy in the use of materials and supplies; the reading of magazines and books pertaining to the engineering profession, the posting of suitable mottoes in conspicuous places, all help to make him a better man.

Some time in life every man bumps into his "stone wall." The real man lands on the other side—Think it over. Be not overcome by difficulties, but overcome difficulties with an effort born of determination to win.

I have seen men go home from their work in the power plant wearing dirty, greasy overclothes and with hands and face unwashed. I have gone into the plant where these same men worked and found the appearance of things there untidy also. Unfortunately—for these men and for the profession—engineering magazines, with their helpful suggestions, valuable information and inspiring influence, do not often find their way into such engine rooms. If they did, conditions would soon be changed. A neat appearance will go far toward making a man think better of himself and of his work.

C. D. ELDREDGE.

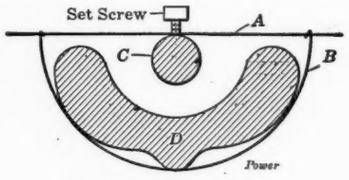
Fairport Harbor, O.

Piston Rod Clamp

The piston rod of some types of pump is often ruined by using a pipe wrench to hold the rod when removing the jam nuts in the water cylinder when about to pack the plunger. After the piston rod becomes badly marked it is next to impossible to keep the stuffing boxes tight.

A device that will save the piston rod a great deal is shown in the illustration.

The piece *A* is made of soft steel, 16 inches long, 1 inch wide and ½ inch thick. *B* is a piece of soft steel of ½



PISTON-ROD CLAMP

inch diameter and about 2 feet 4 inches long, bent so as to fit around the body piece *D* of the pump. *C* is the piston rod. The ends of *B* hook over the ends of *A*, which is slotted. To prevent the piston rod from turning, it is only necessary to tighten the set screw.

ALFRED WOOLCOCK.

Evelette, Minn.

Manholes in Boilers

Although boilers have been manufactured and used for years, there are many still made and installed where little attention is given to the very important feature of accessibility, and this applies both to the setting and the boiler proper.

It is safe to say that a number of disastrous boiler explosions can be partly or wholly traceable to the fact that the design of the boiler and setting prevented proper cleaning and inspection.

But a few years ago it was the custom of some manufacturers to put handholes in the bottom of the front and rear heads of horizontal return-tubular boilers, and these have undoubtedly been the indirect cause of many a bag, blister, fracture and burned blowoff pipe.

The handhole in the back head is usually a constant source of trouble. Of late years the majority of boilermakers are putting a manhole in the bottom of the front head and no hole at all in the bottom of the rear head. This is an improvement over the handholes, as it permits of proper cleaning and inspecting. Usually the manhole in the bottom of the front head is 10x14 inches and, while it is possible for a good-sized man to get through a hole of this size, it would be much easier if it were 11x15 inches, and in most boilers the larger size could be put in with no additional cost and without weakening the boiler head. Boilers made by a certain firm have crow-feet braces, so arranged that it is necessary for a person to make a quarter turn of the body, after getting through the manhole, so that he can drop down between the braces to the top of the tubes.

In nearly all uptodate water-tube boilers, the openings are ample, but in one type the rear drum is so obstructed by a large mud pan that it is almost impossible to get into it, or to do any cleaning.

The openings in the settings of water-tube and horizontal tubular boilers are of all shapes and sizes, but in most of the former type they are ample for the purpose. One manufacturer, however, furnishes castings with 14-inch round holes, which are about the worst thing that could be installed. The brick is laid in the wall to conform with the casting and the result is an opening that is very difficult to get through, or to hoe out ashes and dirt. Square holes about 14x18 inches would insure better care and inspection of this type of boiler.

Every prospective buyer of a boiler should insist that the manholes be made 11x15 inches, that the through and crow-foot braces be arranged to permit easy access, that the openings to the combustion chamber be at least 15 inches wide by 24 inches high, or larger, and so placed that ashes can be readily removed.

THOMAS J. HANNA.

Cincinnati, O.

Making a Low Pressure Trap

The engineers in a certain plant were kept busy running around blowing the bypass on the high-pressure traps in order to keep the pipes drained.

This trouble was due to the traps being worked at too high a pressure, causing the pot in each to hang up to its seat. Some of these traps were remedied by putting in pressure-reducing valves where the high pressure was not needed. The remaining traps were made suitable for high pressure by plugging the seats and drilling a hole through the plug one-half the diameter of the original opening, thus reducing the area of the seat against which the valve was held by the steam pressure. An automatic air valve was put on the cover of the traps and the engineers then forgot all about them.

W. T. MEINZER.

Brooklyn, N. Y.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Driving Keys

With reference to Mr. Taylor's letter on the above subject, which appeared in the issue of November 22, I believe that a few friendly criticisms will be beneficial.

Where is the necessity of marking or measuring the key at all, preparatory to driving it? The object to be attained is to take up whatever lost motion there happens to be between the pin and the brasses regardless of how far the key must be driven. He says that in driving a key the first time, in order to determine the proper degree of tightness, the connections should be moved sidewise or, if this is impossible, to drive the key solid and then back out the required amount. This is correct and the only safe method of performing that operation at any time; therefore, marking the key is of no benefit whatever unless when backing out after being driven solid. I fail to understand how anyone can determine accurately, as he says, just how far a key should be driven each time, as that is equivalent to knowing the exact amount of wear that has taken place, which is out of the question.

Again, he says that the wear is continually making the connecting rod shorter and putting in liners carries it back. From what he states about inserting liners on both sides of the pin he would have us believe by the insertion of liners on the key side we could rectify the length of the rod. A liner thus inserted, with the key remaining the same, would have the same effect as driving the key, thus making matters worse. If the key is on the side of the crank pin next to the cylinder the wear and consequent keying up will diminish the clearance in the head end of the cylinder. The same effect will be produced with reference to the crosshead pin if it be keyed on the side nearest the connecting rod.

In order to equalize the clearance, liners must be inserted behind the brasses on the side of the pins opposite the keys, or all may be put behind either one of them. This is assuming the piston rod to be keyed in the crosshead. If it be screwed in, the liners are unnecessary unless the keys have been driven clear up.

When the crank-pin key is driven the connecting rod moves while the crank pin remains stationary and in driving the crosshead key the connecting rod remains stationary and the pin moves.

The connecting rod may be assumed to be a long box. If we first key the crank-pin brasses to take up the lost motion, the connecting rod moves toward the head end, carrying everything with it except the crank pin and its inner brass. Then if the crosshead key is driven the pin will move still further toward the head end, forcing the piston along with it.

An engine with the crank-pin key or adjusting wedge on the opposite side of the pin from the connecting rod, and the crosshead key next to the rod, has a tendency to keep the clearance equal as the wear on one pin and set of brasses will offset that on the other. A great many engines are so built but the wear is rarely the same on both.

JOSEPH STEWART.

Hamilton, O.

Boiler Tube Failures

In the issue of November 29 there is an editorial, headed "Boiler Tubes," referring to some remarks made by Charles S. Blake, of the Hartford Steam Boiler Inspection and Insurance Company, regarding tube failures in recent times and citing one instance in particular, which is illustrated and described on page 2131 of the same issue.

It is hardly necessary to make any comment on Mr. Blake's contention that the thicknesses used today are identical with those used 25 years ago, whereas the pressure is now 80 to 100 pounds greater, for it is possible to obtain any gage required simply by specifying. Moreover, in those days charcoal iron was the only material used. Now we have hot- and cold-drawn seamless and lap-welded steel tubes, which, for the same gage, are nearly twice as strong as the charcoal iron under internal pressure, so that under the same conditions, by the use of a tube made of soft steel, twice the factor of safety may be obtained.

In the case of failure cited, it is clear that the low factor of safety was largely responsible for the trouble. The tube is reported as being of charcoal iron, No. 10 gage, which had been thinned down by wear to 1/32 of an inch at the point of initial rupture. Besides being nearly twice as strong for the same thickness, as well as more ductile, steel wears better and will not thin down so quickly as charcoal iron. Referring to failures of this kind the editorial states, "There is something wrong with the making of a tube which fails in that way." If charcoal iron is preferred for boiler tubes which are to be used under modern conditions, would it not be only reasonable, in order to maintain the same factor of safety as was provided for under former conditions, to increase the thickness proportionately?

F. N. SPELLER.

Pittsburg, Penn.

With Consulting Engineer Assisting

Mr. Weaver, in the November 29 issue, says, "operating engineers frequently say 'the engineer in charge is best qualified to pass on contemplated improvements.'" Well, it is my opinion that he *should* be, but it depends largely on the man whether he *is* or not. Every operating engineer should keep himself so well posted on the best modern practice that he could take intelligently in hand the rebuilding of his plant, or any part of it, at any time that it might be desirable.

How can he best do this? By carefully studying the *advertisements* that appear from week to week in such periodicals as POWER, and noting any item that would be of benefit in his plant, then writing to the makers for printed matter concerning these articles, carefully studying the literature thus obtained and filing it for use when needed. Then, when the time comes to make alterations and improvements, he is the man who should have the last word. A splendid source of valuable information in regard to any particular department will be found in the catalogs of the manufacturers of goods used in that department.

I have many times been saved from making errors, and have received information of much benefit to myself and my employers, by sitting down and talking with an intelligent salesman, and one rarely leaves me without first giving me some practical information to add to my store of knowledge.

However, the consulting engineer, from his broad experience in many plants, is not to be despised as an *assistant* to the operating engineer, and as such can be of great value; but the man that must

operate the plant, perhaps 365 days in the year, should have the matter entirely in his hands from beginning to end, and his word should be final.

E. H. ROBERTS.

Norwalk, Conn.

What Causes the Engine to Run?

Referring to Mr. Teer's letter in the November 1 issue under the above caption, the following may answer his question:

Indirect, balance slide-valve engines take steam at the center of the valve instead of at the ends as in the direct, balance slide-valve types. When the bleeder valve on Mr. Teer's engine is opened, the steam passes through the bleeder pipe and enters the cylinder at both ends through the cylinder cocks. Ordinarily, the same pressure acts at both ends of the cylinder. But, if the engine is in the starting position, the steam that enters the head end through the cylinder cock cannot escape because the exhaust port is closed. Therefore, there is enough greater pressure in the head end to start the engine to turning over while the steam that enters the crank end through the cylinder cock escapes through the exhaust port until the engine has turned far enough to close this port for compression. Then, the momentum of the flywheel will carry the piston past the crank-end center when the exhaust port on the head end opens and lets the pressure drop. The crank-end exhaust port now being closed the steam cannot escape; therefore, there is enough pressure in this end to keep the engine turning over, and the momentum of the flywheel carries the engine over the center each time. Thus, the engine will continue to run as long as it gets steam in this manner.

ROBERT H. DUNLAP.

Syracuse, N. Y.

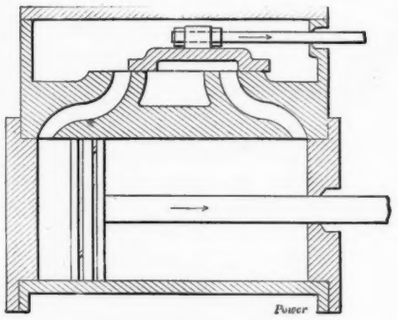
In the issue of November 1, E. R. Teer has a letter under the above title.

Referring to Mr. Teer's sketch it will be seen that with valve *A* open the steam is admitted to both ends of the cylinder through the drain cocks at either end of the cylinder as well as to the exhaust pipe. The reason that the engine will run is as follows:

The steam enters at both sides of the piston, but the pressure of the steam is not the same on both sides, as will be seen by a study of the accompanying figure.

The steam which passes up into the cylinder at the head end cannot get out but simply fills the steam chest, while at the crank end the steam passes up into the cylinder as before, but as the valve has moved almost to the end of its stroke to the right, thus opening the port to the exhaust cavity, the steam escapes through the exhaust pipe.

It is well known that there can be no flow unless there is a drop in pressure. Thus, there is a drop in pressure as the steam escapes through the exhaust port and through to the exhaust cavity at the crank end, while at the head end there is no flow and consequently no decrease in pressure. In this way a greater pressure is brought to bear on the head end, and if it is sufficient it will run the engine.



SECTION THROUGH CYLINDER AND VALVE CHEST

If, however, there is too much friction, there will not be sufficient pressure to run the engine.

E. S. LIBBY.

Chicago, Ill.

Power Plant Design and the Operating Engineer

In the November 29 issue I read Mr. Weaver's contribution under the above heading and, while the consulting engineer is appreciated if he is a good one, we must take Mr. Weaver's attack on the operating engineer as rather unwarranted. He said, in part, that every day one sees mistakes in the layout of power plants, owing to the designer being thick headed. The natural question is, who sees these mistakes? The answer is the operating engineer. Why? Because if he is an engineer of practical experience and technical knowledge, as every operating engineer should be, he has operated other plants and knows how he would have designed this particular plant to obviate the mistakes. Mr. Weaver further says that he believes that in the majority of cases blunders in power-plant design are due more to self inflation than to any other cause. I am glad that I can concur with him in this statement. As a rule this self-inflation is found in the inexperienced rather than in the mature and experienced engineers.

Mr. Weaver tells us that the consulting engineer laughs up his sleeve over mistakes made by the operating engineer due solely to ignorance of the laws of philosophy, simple laws which everyone should know. Perhaps Mr. Weaver has in mind starters-and-stoppers or oilers. The operating engineer can buy and read any engineering work published, providing he has the price. The engineering magazines keep him up to date. He may not be as good a draftsman or as conversant with the higher mathematics as the graduate of the school of technology, but he certainly has every means and method of obtaining engineering data that anyone has. And it is owing to this knowledge combined with practical experience that makes the operating engineer able to bring some semblance of order out of the chaos left him by some designing engineers.

Again, how can the engineer in charge secure uninterrupted and satisfactory service while watching the hundred and one things around a power plant, continually looking for places where improvements can be made and at the same time be a designing engineer? To begin with, if the engineer in charge could have the designing of his plant, he would not have a hundred and one places about his plant where improvements are needed and a smaller per cent. of his time would be needed to secure uninterrupted and satisfactory service. If he has had charge of a plant for some time and does not know where the improvements are needed he is a very ignorant or lazy man and should be replaced at once by an *engineer*. Will Mr. Weaver tell us how a man can properly design a steam plant who has not had a wide operating experience? Would he take swimming lessons of a man who had never been in the water above his knees? Would he employ a doctor of medicine who was just graduated and had no hospital or other experience? I think not. And no man should call himself a consulting engineer, no matter what his educational advantages may be, until he has had at least ten years experience operating steam plants. This is the kind of consulting engineer that is needed; men who are not blinded by preconceived ideas. Talk about the mistakes of the operating engineer, an issue of POWER could be filled several times with details of the mistakes made by designing engineers. The operating engineer does not laugh up his sleeve or in any other way; he has to get busy and reconstruct and correct as far as possible their mistakes.

Mr. Weaver, like many other writers, uses the term *operating* engineer or chief *operating* engineer. In Webster's dictionary we find that an operative is a laboring man, a laborer, artisan or workman in a manufactory. The engineering papers and the men in charge of steam plants have been trying to make steam engineering a profession. No wonder Mr. Weaver thinks that we are on too low a social plane to associate with the self-styled consulting engineer of brains and achievements. In the Massachussetts engineer's and fireman's license laws, section 80, we find that the words, "have

charge" or "in charge" shall designate the person under whose supervision a steam plant is operated. The person operating shall be understood to mean any and all persons who are actually engaged in generating steam in a power boiler. Perhaps Mr. Weaver was thinking of the fireman when he wrote the article under discussion, for no man who could not layout and install a steam plant would be a competent man to "have charge" of it after it was installed.

C. E. BASCOM.

Worcester, Mass.

Does the Crosshead Stop?

We do not know that the editor of POWER who answered the above question on page 1790 of the issue of October 4 needs any defense, but we cannot refrain from arguing the question with H. T. Fryant, whose letter appeared on page 2078 of the issue of November 22.

In Fig. 1, which is a copy of Mr. Fryant's figure, we point out that at the instant that the crosshead reaches the dead point H, the circle of motion of the crank-pin center is tangent to the circle described about H_1 as a center. Therefore, at the instant of dead center the crank-pin center is moving along the outer circle as well as the inner circle. Using, then, the argument obtained from Stevens' "Mechanical Catechism" that,

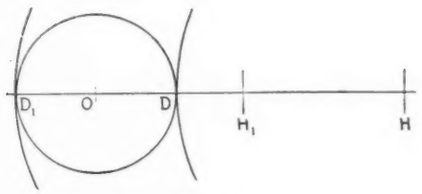


FIG. 1.

"The crosshead center could stand still only if the crank pin moved around it as the center," we have proved that the crosshead center does stand still at one instant. A similar argument shows this to be true also for the point H.

We submit, further, the following proofs, which may be more rigorous. In Fig. 2 let V represent the absolute velocity of the crank-pin center at any instant, being drawn to the point shown for the sake of clearness only. Resolving this velocity into its components perpendicular to and parallel to the motion of the crosshead-pin center, we represent the velocity parallel to the motion of the crosshead by Vx and perpendicular to it by Vy. Let be the angle between V and Vx. Then,

$$Vx = V \cos. \theta$$

at any instant. Vx equals zero when θ equals 90 degrees. At the instant that the crank-pin center crosses the line of motion of the crosshead-pin center, θ equals 90 degrees and $\cos. \theta$ equals zero. Therefore, Vx equals zero and the motion of the crosshead-pin center is zero. An exactly analogous argument is used for either point H or H_1.

In Fig. 3 is presented a third proof which we consider equally good with that connected with Fig. 2. Let ds be

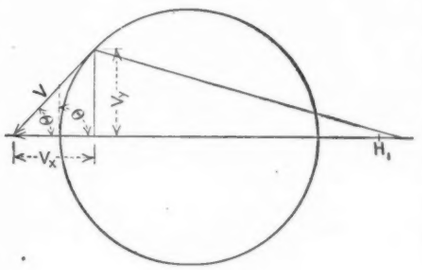


FIG. 2.

the infinitesimal distance traveled in a time dt. Then $\frac{ds}{dt}$ equals the velocity; that is,

$$\frac{Space}{Time} = Velocity$$

This distance ds may be traveled along the path dp perpendicular to the axis of the connecting rod, and then along dm parallel to the rod. The quantities $\frac{dp}{dt}$ and $\frac{dm}{dt}$ are, then, the respective components of the velocity $\frac{ds}{dt}$. Let ϕ be the angle between the axis of the connecting rod and the line of motion of the crosshead-pin center. The angle between ds and dp is a function of ϕ, say ϕ', by the geometry of the figure. Then, dm equals ds sin. ϕ'. When ϕ becomes zero at the point H_1 or H, sin. ϕ equals zero, and then dm equals zero, or the crank-pin center has no motion parallel to the axis of the connecting rod; then, since the crosshead-pin center has no motion perpendicular to its line of action and since it can have at that instant no parallel motion, it must stop.

We do not doubt but that there is no

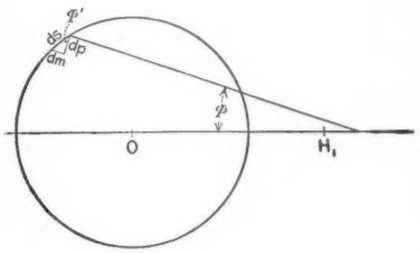


FIG. 3.

interval "between the ending of one day, or month or year and the beginning of the next," but this has no bearing on the motion of a crosshead or on the pendulum motion of a clock. The clock merely serves as an indicator of the passage of time, and as such is necessarily subject to mechanical limitations which time is not. The pendulum of a clock does stop at either end of its arc, for at these points the kinetic energy of the pendulum mass, or its energy of motion, is all converted into potential energy or energy of position, which must then mean that the pendulum stops.

We have shown that the crosshead does stop, and, that being the case, it does stand still, no matter how brief that time may be. The crosshead does not change direction without stopping. After the instant of stopping, say at the point H_1, the angles θ for Fig. 2 or ϕ for Fig. 3 become opposite in sign and there is an increase in the velocity component parallel to the axis of motion for the crosshead-pin center for Fig. 2, or parallel to the axis of the connecting rod in Fig. 3. The mathematics of this need not be discussed.

If there is anyone who still fails to see that the crosshead stops, we would like to have him show that a railroad train does not stop before it backs up.

C. R. STOVER and M. W. PULLEN.

Boston, Mass.

The Economy of Vigilance in Small Matters

A correspondent in the issue of December 13 refers to losses due to small leaks. While this matter has been commented upon quite often, it should be kept continually before the eyes and drilled into the mind of every engineer.

There are several hundred valves and compression cocks, a large laundry, etc., in the building where I am at present employed. When I started to work here, some years ago, many of these valves and cocks were leaking and water was being wasted in other ways. By repairing all of the valves, and by continually requesting employees not to needlessly waste the water, the consumption was reduced by between fifteen and twenty thousand gallons every 24 hours. As much of the water which was being wasted was hot, and as many of the other leaks were steam, this saving meant also a saving in fuel.

As the men who deliver the coal to the boiler rooms and other parts of the building have an abundance of spare time, I got them to sift all of the ashes. During the first year that this was tried the fuel consumption was reduced by about 100 tons. This reduction has been kept up year after year. If extra help had been required to do the sifting, it would not have been a paying scheme for one man's wages would have been more than enough to eat up the amount of money saved in the coal bill.

During eight years the only extra cost charged to this particular item has been $30 for repairs to the ash sifter.

JAMES E. NOBLE.

Toronto, Ont.

Leakage through a Piston Valve

I was extremely interested in the publication in a recent number of results of tests made by George Mitchell, in the testing laboratory of the University of Pennsylvania, on the leakage of piston valves under actual operative conditions. I have made a great many visual tests myself on various forms of valves to determine the presence of leakage, but not to determine the actual amount. In these tests the method of showing this leakage was to set the valve in the center of travel, where it covered both ports; the exhaust valve under the engine in the exhaust line and cylinder drips were closed, the indicator plugs at both crank and head end of the engine were removed, and the throttle valve opened. In not one of over fifty of these tests for leakage of piston valves and flat pressure-plate valves was it practicable to open the throttle full for the reason that the leakage as shown by the steam escaping from the openings caused by the removal of the indicator plugs was so excessive that the room was immediately filled with steam.

Such a test naturally had to be made while the engine was in a state of rest.

The argument was often made that these tests were not fair ones, so far as the engine was concerned, inasmuch as a film of water between the valve and the bushing in a piston valve, and between the valve and the pressure plate in the pressure-plate valve, effectually eliminated the major portion of the leakage that was shown by these tests when the engine was not in operation.

The test, therefore, by Mr. Mitchell conclusively proves that the argument of water filling these spaces is a fallacy, as in his test the engine was operated under normal conditions.

The argument will probably be made that inasmuch as this valve was not equipped with packing rings, the leakage was greater than it would have been on a valve so equipped, but from the tests I have made I cannot notice any difference in the amount of steam escaping through valves equipped or not equipped with rings.

We have all heard the argument that the rings in a piston do not show excessive leakage, therefore why should the rings in a piston valve show any more leakage? The answer to this argument is simple:

If an engineer discovers a flaw or blow hole in the barrel of his cylinder, he would condemn the cylinder, for the reason that the piston rings traveling across this flaw would soon be cut and cause leakage. The rings in a piston do not travel across any ports, but merely up to the counterbore, whereas the rings in a piston valve must travel across the ports, and the ports are usually designed with bridges to prevent the rings from falling into them. The spaces between these bridges accomplish the same result, only to a much greater extent, as the small flaw in the cylinder would, for the reason that the bridges do not have as much bearing surface as the full bore of the valve seat and consequently wear faster. This greater wear at this point causes the rings to move in and out of the valve when crossing the ports, causing excessive wear on the rings, both on their circumference and on the side fit in the grooves.

I believe a test was made at Cornell University several years ago on a piston valve equipped with rings which could be tightened by hand, and I understand from one of those present that it was found that in three and a half hours after starting, the leakage was so great that the engine had to be shut down and the rings reëxpanded by hand.

I note a letter published in POWER for November 8, by A. L. Ide & Sons, in which they give the results of several interesting tests to determine the increase in steam consumption with piston valves made 0.01 inch smaller than what they term "commercial fit." With 150 pounds steam pressure they got a consumption ranging from 26 to 27 pounds with valves with commercial fit. They then tested two valves that were made 0.01 inch under size and found that the steam consumption was increased in one case to 32.7 pounds. Taking an average of the results obtained with valves of commercial fit at 26½ pounds, it will be seen from these tests that a valve 0.01 inch smaller increased the steam consumption 22.3 per cent., which confirms Mr. Mitchell's findings of 22 per cent.

I think that any engineer will find on measurement that a piston valve, if it has been operating at least a year, will be even more than 0.01 inch smaller than the bore; in fact, I recently made a test of a piston-valve engine, the steam consumption of which was 54 pounds per brake horsepower per hour. I measured the valve and found it to be over 0.03 inch, or to be exact, 0.033 inch smaller in diameter than the bore.

Taking the calculations given by A. L. Ide & Sons, that each 0.01-inch wear increased the steam consumption 22.3 per cent., and assuming that the steam consumption of the engine I tested was 30 pounds per horsepower per hour when new, the leakage through the valve on this engine would be at least 3.3 times 22.3 per cent., causing an increase in steam consumption of 73.6 per cent. Therefore, if this engine with a tight valve would develop a horsepower-hour on 30 pounds of steam, 73.6 per cent. increase would result in a total steam consumption of 52 pounds, which is a trifle less than that indicated by the careful test which I made on the piston-valve engine.

I do not agree with the statement of A. L. Ide & Sons that a valve 0.002 to 0.003 of an inch under size is plainly a poor fit. The sliding fit for a hub on a shaft is from 0.002 to 0.003 inch for ordinary diameters, when the parts are cold, and I maintain that any piston valve must be at least 0.002 to 0.003 of an inch smaller than the bore to be free to slide.

From the visual tests I have made, the leakage on all flat-valve engines equipped with a pressure plate has been greater than on the piston valve. I say greater, for the reason that a greater amount of steam is always seen escaping through the openings left by the removal of the indicator plugs.

I have surprised a great many engineers by making this statement and the result has been that quite a number of these tests have been made, all of which confirmed my claims. Other confirmation may be had from the tests made by Messrs. Dean & Wood, the results of which were presented in a paper to the American Society of Mechanical Engineers at the meeting in Detroit, in 1908, an abstract of which, I believe, was published in POWER. These tests also showed the flat pressure-plate valve to be a very leaky device.

I happen to know of a very fair test that was made on a pressure-plate valve engine in New York City recently, the results showing a steam consumption as high as 59 pounds per indicated horsepower per hour. All of which helps to prove the point that I am trying to make, viz., that a balanced pressure-plate valve shows greater leakage after a certain period of operation than the piston valve. This is probably due to the fact that the pressure-plate valve when absolutely new leaves the builder's factory with 0.003 to 0.004 of an inch clearance between seat and pressure plate. This clearance has been found necessary in order to provide freedom of action under all conditions, and this clearance becomes greater with use. Mr. Mitchell's test was made on a valve that was new, and had not been running for any considerable length of time, and I know from experience that this leakage would be much greater after the valve had been in operation for several weeks. According to the Dean & Wood report, valves can, and are, being made that are self-expanding to compensate for wear, and thereby eliminate leakage.

I wish Mr. Mitchell would make another leakage test on this valve after it had been run for four weeks' time, and also make a leakage test on a flat pres-

sure-plate valve, when new and also after it has been in operation for a short time. These are just the tests that the engineering fraternity has been wanting for years, and Mr. Mitchell is to be complimented on his manner of making these tests, which method disposes effectually of any arguments that have been brought forth that both types of valves above mentioned do not leak steam under operative conditions.

F. A. SHOEMAKER.

Buffalo, N. Y.

Selection and Use of Packing

Engineers have but few subjects to consider that are capable of greater diversity of opinion than that of the results obtained from the various piston-rod packings now in general use. However good or however worthless they may be when considered separately, the results that are obtained are so conflicting that what is considered satisfactory by one is as honestly condemned by another. The great variety of conditions under which packings are used is, and always will be, understood as unavoidable. How to meet these different requirements could be answered by a hundred engineers, in a hundred different ways, according, of course, to the various experiences of each one, those experiences being due, of course, to the different conditions met with in each case, such as speed of piston, pressure, material, etc.

There are, however, faults attributable in many cases not only to the engineers, but also to the manufacturers and inventors of these articles of everyday use. It would be an injustice to assert that every engineer does not use intelligence in the matter of using an article. Yet it is often found that a well made article is condemned and thrown aside, owing to a want of knowledge as to its proper use or to carelessness in application. At the present time there is an abundance of varieties of packing, each one, of course, claiming a certain superiority over the rest. It is, nevertheless, true that in many cases quality has been sacrificed to price, notwithstanding the fact that manufacturers stoutly deny any difference in the quality of their product. Is it reasonable to suppose that such an assertion is true when one can buy at the present time for 50 cents an article similar in appearance to what formerly cost three times as much? I think not. This reduction in quality is often done in such an expert manner, however, that none but experts in fibers can detect it from outward appearance, and thus in numerous cases the poor results are attributed to some accident or carelessness in using.

This condition of affairs can only be remedied by the men who use the packing, and who should insist upon obtaining an article of high quality suitable to their own particular case, whatever the cost may be. The compensation will be obtained in the saving of wear and tear, fuel, power and in repairs. Packings having a flat surface against the rod allow the most perfect bearing and produce the least friction as the pressure is more uniformly distributed. Avoid those packings which produce a hard or rigid bearing beyond what is necessary to prevent leaking, which should be provided for by natural expansion. Of course, I assume that the rod is perfectly level and in line; trying to hold a rod steady by screwing up the packing hard when everything is out of line usually results in a miserable failure. If metallic packings are used the conditions must be more perfect to insure success than is generally the case with fibrous packings. However, metallic packings are now being used with good success when applied intelligently and, when they are proof from the efforts of the monkey-wrench engineer, are giving excellent satisfaction. I have observed cases where metal packing was in use where when the least little moisture occurred around the rod, due to its not being exactly in line, or to a poor adjustment of the crosshead, the engineer screwed up on the nuts of the stuffing box until fire appeared. Men of this class will ruin any kind of packing, but it is a pleasure to say that they are very much in the minority. I do not presume to enumerate either the ordinary or the desirable makes of these goods, believing that these few hints may serve to arouse interest in the packing problem and thereby be instructive to engineers in general.

CHARLES H. TAYLOR.

Bridgeport, Conn.

Hotel Power Costs

In the issue of POWER for November 8, under the head "Hotel Power Costs," I saw the figures given by J. H. Morrow, chief engineer of the Great Northern hotel, Chicago. Mr. Morrow gave definite figures as to the power costs for a year before and after installing an isolated plant. I would be very glad to hear from other engineers of hotel and office buildings along this same line. Also, I should like to secure some data as to the cost of an installation per kilowatt.

Which would be the more economical for a small plant, a steam turbine or a reciprocating engine driving a generator where the exhaust is to be used in a heating system?

O. L. SHERMAN.

Duluth, Minn.

Introducing Solvents into Boilers

The article by C. H. Taylor in the December 6 issue on this subject will help quite a number of engineers who, like myself, have looked vainly through the advertising pages of POWER for a thoroughly reliable compound feeder. In several plants that I have visited there is some homemade affair that is doing the work with more or less success. Some time ago I went so far as to write a large manufacturing concern whose business was among other things, the making of force-feed lubricators, asking if they did not believe they could put something on the market that would "fill the bill," constructed on the lines of a force-feed lubricator which could be attached to the boiler-feed pump. Some compounds may have a sediment, thicken up, or contain acid which might attack the metal of the pump or valves. If any of the readers are using an ordinary force-feed pump with success, I would be glad to know it. There are quite a few little difficulties which crop up in connection with compound feeding.

Leakage at the gland would be particularly objectionable; hence a well designed stuffing box is essential. The tank or vessel from which the pump takes its supply should be elevated and have a graduated gage glass. The quantity pumped should be easily adjusted. An apparatus of this kind ought to command a ready sale. The amount of compound introduced would have a direct relation to the quantity of feed water put into the boiler, and this is a desirable condition. I have an apparatus that is working in a fairly satisfactory manner and which I partly made from scrap. The homemade product, however, is usually very crude, clumsy and quite frequently more costly than the originator would care to admit. There is so much of this homemade stuff to be found, so many ingenious, temporary repairs, etc., which, in the long run, are apt to prove extremely costly that anything of this nature ought to receive very careful consideration before adoption.

FRANK H. WILLIAMS.

New Haven, Conn.

The Problem of Smoke Abatement

Some time ago POWER published a paper by Mr. Randall, on the suppression of smoke. Mr. Randall's paper was very interesting and readable, but he gave no real solution to the smoke problem.

As things stand at present, particularly as regards the payment of the firemen, no device made is really satisfactory in the suppression of smoke. This is no fault of the device in many cases; it is simply lack of knowledge on the fireman's part as to the method of utilizing the device to do away with the smoke. Steam jets, special arches, special grates and special furnaces and all characters require expert handling in order to obtain results, and they rarely get it. An expert fireman can get an almost smokeless fur-

nace, unless it is being forced to the limit, with almost any of the coal sold in New England. If, however, an inexpert fireman takes the shovel, the results are radically different, and no matter how good the furnace is, smoke will be given off in large quantities.

The majority of manufacturers look on the fireman as merely a device for feeding coal under the boilers. Brute strength is the main consideration, and brains receive but scant acknowledgment. This seems rather extraordinary when one considers that one of the heaviest expenses in the power plant is the coal pile. A manufacturer will employ a first-rate engineer and pay him a good salary, for the express purpose of keeping his engine and shafting in good condition. He will purchase and install a high-grade engine and condensing outfit, if possible, in order to economize in steam consumption, but when it comes to the man who is largely responsible for the amount of coal burned on the grate, he is always looking for the strongest man at the lowest price. At this very point there is a tremendous opportunity for saving. A good fireman is worth money, and he will more than save the difference in his wages through the use of his brains. Less coal will be required and less smoke will be made. Less coal will be required, because he will see that the amount of air admitted to the furnace is graded to the amount of coal he puts on the fire, so far as the construction of the furnace will allow, and, therefore, he will burn up the volatile matter in the coal to a very large extent, which under the firing of a poor man, goes up the stack in smoke.

There is, of course, vast room for improvement in the construction of the furnace at present used for bituminous coal. In the first place, the hight of the boiler above the grate is far too small. There is not a sufficient opportunity for varying the amount of air which enters above the grate. The ashpit door is not large enough to give a thoroughly good admission of air under the grates, and the greater part of the air goes up through the grates at the back of the furnace, instead of at the front. All engineers know the material abatement in the smoke nuisance that can be secured through the use of the dutch oven and the very long fire box. This is due mainly to the thorough mixing of the air and volatile gases and the opportunity offered for them to ignite before they go over the bridgewall. In order to burn these gases, it is necessary that they be mixed with a proper amount of air and brought to a sufficient temperature to be ignited when they pass over the rear part of the fire. If they are not ignited before they go over the bridgewall, there is little or no prospect for their being ignited at all. Special arches in many cases will improve the combustion to the suppression of smoke, and increase the efficiency of the combustion.

My solution of the smoke problem is, then, pay the fireman enough to make it worth while his staying on the job, and then train him. Adjust the furnace so that air can be admitted both above and below the grate and thoroughly mixed with the combustible gases before these gases go over the hottest part of the fire. Train the fireman to coke the coal at the front of the fire, so as to drive off the volatile matter from the front end of the furnace. Push this coal back as the fire burns out at the rear, and shovel again in the front. It might be said that the automatic stokers do away to a considerable extent with the skill required by fireman. This, however, is by no means true. The automatic stoker requires adjustment of the air just as much as does hand firing; and a poor fireman cannot do as well with an automatic stoker as a good one.

Boston, Mass. HENRY D. JACKSON.

Air Bleeder for Boilers

In the December 13 number, Mr. Mistele has a letter on the subject of water hammer, in which he speaks of the trouble he experienced with air in the boilers when raising steam and the remedy he applied.

I regard it as a wise plan to have all boilers tapped at their highest point with a 1-inch bleeder connection. In raising steam this should always be left open until the gage shows some pressure. It should be given a good, strong blow before opening the stop valve to the line; particularly is this the case where condensing engines are being run, as a boiler which contains very much air will frequently cause a condenser to go down unless it is thoroughly drained of air before being cut in.

Another desirable feature of such a bleeder is that it allows a much more rapid cooling of the boiler when a hurry-up job of washing out is in order, as is so frequently the case in small plants where but a few hours can be had in which to cool down and wash a boiler.

O. B. CRITCHLOW.

Woodlawn, Penn.

Faulty Design

Under the above caption, Mr. Rayburn, in the November 22 issue of POWER, describes a condition of engineering which is not engineering; an exception is taken to his ruling of the term "engineer." The manifold errors cited in the installation of this particular steam plant tend to prove that an engineer, a real engineer, was not in evidence—degradation should not be cast upon the engineering profession at large to style those who were engaged as "engineers." As in all walks of life, in the engineering business today there are engineers, and there are engineers—there is a wide distinction. The "catalog" engineer, the inexperienced engineer, the inefficient engineer, the engineer who could not fill a drafting position, who follows the calling haphazard, and subsists on the earnings derived from the enterprise promoter not familiar with such design, who accepts a fee from his client and an additional fee from some particular manufacturer whose product he specifies and insists upon, these are not engineers, nor are their efforts engineering—it cannot be classed in that category.

The real engineer, the man who knows his business, does not necessarily have "to smoke black cigars and carry a slide rule," knowledge is not contained in these two elements, and I am under the impression that some engineers use neither. The real engineer is not given to words or boasting; as a rule, he is open to reasonable argument, and his actions and methods show results.

The selection of a consulting engineer for a certain work shouid be made only after investigation as to ability and past performances; the real engineer is alive to the best interests of his client, he expects future business from him, he expects his recommendation to others. The competent engineer always proves to be the cheapest in the long run.

L. R. W. ALLISON.

Los Angeles, Cal.

Water Gage Connections

In the December 13 issue of POWER, Mr. McGahey has a letter relating to the placing of valves in water-column connections. The pros and cons of this matter have been discussed in the columns of the mechanical papers and outside of them many times, but I do not remember ever having seen stated what I regard as strong justification for their use; namely, their great value when the water connection becomes clogged. With a valve in the steam connection which can be closed, full boiler pressure can then be brought to bear to blow out any obstruction in the pipe, whereas, with no valve in the steam pipe, the opening of the blowoff valve on the column but imperfectly cleans the water connection between the column and the boiler, due to the fact that the pressure in the column is, to a great extent, balanced. The same principle applies here as in the case when we close the lower valve first when a gage glass breaks, allowing the steam valve to blow and thus hold back the hot water. I was once saved from the necessity of a shutdown at a critical time by having these valves to use, and later when a Hartford inspector recommended their removal, I was able to convince him that it was better to leave them in.

O. B. CRITCHLOW.

Woodlawn, Penn.

Steam Turbines and Generators

By E. D. Dickinson
and L. T. Robinson

A paper presented at the December meeting of the American Institute of Electrical Engineers, taking up the subject of testing turbo-generators from the point of over-all efficiency under conditions which are as near as possible to those under which the machine is to operate.

In manufacturing steam turbines a great amount of testing is necessary to determine the effect of changes in design or to verify theories which cannot be established by calculation; much of this is of a laboratory nature. There is also a large amount of testing done to establish the over-all economy of the complete unit, which is all that is of commercial value to those operating steam turbines.

The one positive method of testing a turbo-generator is to measure the steam that goes in through the throttle, and the electrical energy delivered at the terminals of the generator. The surest way of determining how much steam enters the turbine is to collect and weigh all the steam after it has been condensed. This necessitates the use of a surface condenser. In making such a test two things are essential: first, that all the steam used on the turbine be condensed and measured; second, that no steam or water not used in the turbine be allowed to enter the condenser. The condenser should have the leakage checked, before and after each test. This is accomplished by shutting all steam off the turbine and running the condenser for some time with full vacuum, the discharge from the hot-well pump being accurately measured. Split tubes sometimes cause leakage which is difficult to locate as they open only when the condenser is heated by large flows of steam.

When the condensed steam cannot be measured, as is the case when the turbine is operating noncondensing, or when the condenser is of the jet type, the steam consumption is found by weighing the water fed to the boilers. In making such tests, the liability to error is very great, and every precaution must be taken in order that the results may be reliable within a certain degree of accuracy.

The steam piping connecting the boilers and turbine must be disconnected from all other piping, and all openings must be blanked off; valves must not be relied on. All blowoff and drain valves must have their outlets visible, and all piping between the boiler-feed pumps and the boilers must be exposed and have no branches. Leakage of the boiler itself is difficult to locate, as all water or steam escaping is vaporized and carried up the stack. The boiler leakage should be checked before and after each test by closing the throttle to the turbine, or, if necessary, blanking the pipes at the turbine and running a test measuring the amount of water required with full steam pressure on the boilers and piping. The feed water used should be weighed, and not measured by meters.

Tests which have come under my observation have shown a boiler leakage of 10 to 12 per cent., and one particular case showed a leakage of over 20 per cent.

The comparison of efficiencies of different machines is the most satisfactory way of considering their relative merits. To determine the available energy in one pound of steam it is necessary to know the pressure in pounds per square inch, the quality and the temperature of the entering steam; also the pressure at the turbine exhaust. To measure the vacuum a full-length mercury gage should be used. If the steam be superheated, since there is some difference of opinion concerning the specific heat of superheated steam, the figure assumed must be given. In testing turbines consisting of several stages, the pressures in the different stages should be measured, as this affords a check and should show any abnormal conditions which otherwise might not be observed. The kilowatt output should be net, that is, the kilowatts for excitation should be subtracted from the generator output.

All instruments, including meters, gages, thermometers and scales, must be very accurately calibrated before and after the test.

Whenever possible, turbines should be tested under the conditions for which they were built to operate, as corrections for different conditions are liable to cast doubt upon the accuracy of the test. In general, the corrections for steam pressure, moisture or superheat are less liable to be misleading than that for a varying vacuum, as comparatively large changes in any one of the first three will but slightly affect the results, but a slight change in the vacuum makes an enormous change in the available energy.

Electrical Output

First consider the measurement of direct-current output. Usually, station instruments in connection with the generator switchboard have been provided, but unless temperature conditions can be very accurately controlled and the instruments can be checked under operating conditions they should not be used. The station voltmeters may sometimes be satisfactory but it is the usual practice to supply direct-current station ammeters to operate from shunts of approximately 60 millivolt drop, which requires that the indicating part of the ammeter be largely a copper circuit; therefore, the whole combination is subject to considerable error due to variations in room temperature, and with some shunt arrangements, to variations in the current to be measured. For the precise measurement of direct-current output, portable indicating ammeters should be used having 200-millivolt-drop shunts, and permitting the use of indicating millivoltmeters whose circuit consists largely of resistance material having practically no temperature coefficient.

When using either switchboard or portable instruments the influence of any stray fields should be investigated. Also, care should be taken to correct the observed indications of the millivoltmeter for any electromotive forces that may appear in the shunt or leads due to thermo-electric effects. The amount of error due to this cause may be observed by reading the millivoltmeter at the close of the test with no current flowing in the main circuit. Referring again to the station type of shunts, unless the ammeter is checked with the shunt connected into the busbars, care should be taken to see that the distribution of current flow through the shunt is the same when the ammeter is used as when it was tested. It is quite possible to have large errors due to this cause.

Next, it is necessary to consider the proper use of the instrument transformers which usually provide the only means of enlarging the capacity of the instruments to meet ordinary requirements.

In using instrument transformers it is necessary to observe the precaution to have the secondary connected load the same as that which is on the transformers when they were tested for the certificate. It is also necessary if the test is to be made under conditions that will give a low power factor on circuits, to know that the phase displacements in the instrument transformers are not large enough to appreciably affect the results. It is also well not to use instrument transformers with interconnected secondaries except for the common ground which should be employed as a safety precaution.

If possible, a test should be made on noninductive load, in which case, if all the test arrangements have been satisfactorily attended to, the apparent power as shown by the volts and amperes should agree within 1 per cent. with the wattmeter indications and the watts indicated should be taken as the true output. If the test cannot be made at unity power factor, the voltmeters and ammeters should be included so that the general conditions of distribution of load, etc., may be known throughout the test. For this purpose the station instruments would be satisfactory.

Watt-hour meters should never be used unless checked in place at the frequency, voltage, etc., which are to be used in testing. If it is not possible to run a complete test at a fairly steady load it is usually possible to make a few runs on the watt-hour meter under load conditions and to use this check as a basis for determining the output by means of the meters during the test run on an unsteady load. It is still advisable to read the indicating instruments at short intervals so that their indications may be made use of in computing the final result.

Single-phase indicating instruments for polyphase service are to be preferred for precision work to polyphase instruments, for the reason that indications of a polyphase instrument are made up by the two elements in such a way that it is not possible to apply corrections to either element to get the true total result unless the division of load is known by single-phase instruments; and if the single-phase instruments are required for this purpose they may as well be of the precision class and used for the actual determinations, and the polyphase instrument omitted.

Discussion

After the presentation of the paper the discussion was opened and consisted, in part, as follows:

Mr. Dunn: The paper deals principally with over-all efficiency tests, but before these tests become necessary there must be an enormous amount of detail and special testing both of the generator and the turbine. One of the important things to know in regard to both these pieces of apparatus is the proportion of losses chargeable to each.

It is found, for instance, when retardation tests are made on turbo-generators, that vibration, windage, etc., occupy a different proportion of the total losses than on the ordinary classes of apparatus; and the proportion of these losses is not determined by prior conclusions. Empirical methods, only, will bring these out. Consider the mechanical balance of turbo-generators. It is well known that below the first critical speed, if the generator is run in flexible bearings and the chalk held against that part of the revolving surface which seems to be highest, it will hit the heavy part of the revolving member. But when the speed has increased to the point where the apparatus gyrates around its center of gravity then the chalk mark will be moved theoretically 180 degrees from its first position. It may appear easy to calculate these critical speeds but even when the best knowledge on the subjects of vibrations, inertias and gyrations are applied, the result will not agree with that found in practice.

Again, regarding ordinary efficiencies, where the company builds both the generator and the turbine, its responsibility is founded on the amount of steam consumed and the electrical energy developed; but where they are made by separate companies the individual performances are more important.

Mr. Emmet: Individual study of the generator and the turbine is very desirable, but is extremely difficult. This is because the generator is a high-speed piece of apparatus requiring a large amount of power and cannot well be run by anything but the turbine itself. However, there is one method of investigating the generator alone which has considerable value; this is the "deceleration" method. It consists in bringing the generator up to a speed, by motor or otherwise, in excess of that at which it is to be operated and then allowing it to decelerate, noting the rates of deceleration and from these rates, with a carefully calculated moment of inertia, determine the amount of power exerted in deceleration at any particular instant.

A matter of much interest but one which is only slightly alluded to in this paper is that of the steam meter. We have been using steam meters in all of our turbine tests for a long time and at the same time have been weighing the water. The results have checked within 2 per cent. in practically every case.

Mr. Dreyfus: An important characteristic of the steam turbine is that the inlet pressure varies almost directly with the load, provided the same steam pressure, superheat and vacuum are maintained. This affords a means for graphically checking the performance of a turbine.

Clearance in Ammonia Compressors

At the recent meeting of the American Society of Refrigerating Engineers, Thomas Shipley, of the York Manufacturing Company, presented a paper dealing with the effects of clearance in vertical single-acting compressors of the false-head type and horizontal double-acting compressors of the spherical-head type.

The compressors were of the same diameter and stroke and were driven by the same engine, that is, when one compressor was in operation the other was disconnected. The runs were made at suction pressures of 5, 15.67 and 25 pounds gage, and the condensing pressure was 185 pounds gage. The speed was 70 revolutions per minute and during the runs all conditions were kept as nearly constant as possible.

In the single-acting compressor the clearance was controlled by screwing the piston rod into the crosshead and in the double-acting compressor by placing metal rings between the cylinder flanges and the heads.

Table 1 shows the relative effect of clearance on the horsepower per ton, and Table 2 shows the effect on the capacities. It will be noted that the losses due to clearance in the double-acting compressor are much larger than those in the single-acting compressor, and that the losses increase inversely with the suction pressure.

TABLE 1. COMPRESSOR INDICATED HORSEPOWER PER TON.

Linear Clearance, Inch.	Clearance Volume in Per Cent. of Displacement.		5 Pounds Suction Pressure.		15.67 Pounds Suction Pressure.		25 Pounds Suction Pressure.	
	Single-Acting.	Double-Acting.	Single-Acting.	Double-Acting.	Single-Acting.	Double-Acting.	Single-Acting.	Double-Acting.
1/32	0.24	....	1.75	....	1.30	....	1.09	....
1/16	....	0.42	....	2.18	....	1.60	....	1.26
1/8	0.76	0.85	1.77	2.34	1.32	1.62	1.10	1.28
1/4	1.46	1.55	1.81	2.45	1.34	1.64	1.11	1.30
1/2	2.85	2.93	1.82	2.56	1.36	1.72	1.12	1.35
1	5.63	5.71	1.83	2.89	1.39	2.01	1.13	1.44

TABLE 2. TONNAGE PER 24 HOURS.

Linear Clearance, Inch,	Clearance Volume in Per Cent. of Displacement.		5 Pounds Suction Pressure.		15.67 Pounds Suction Pressure.		25 Pounds Suction Pressure.	
	Single-Acting.	Double-Acting.	Single-Acting.	Double-Acting.	Single-Acting.	Double-Acting.	Single-Acting.	Double-Acting.
1/32	0.24	....	2.27	....	38.0	....	50.4	....
1/16	....	0.42	....	19.2	....	33.0	....	47.4
1/8	0.76	0.85	22.6	17.3	37.4	32.1	50.1	45.1
1/4	1.46	1.55	21.0	16.0	35.6	30.0	49.1	44.8
1/2	2.85	2.93	19.7	14.3	34.4	28.9	47.0	42.3
1	5.63	5.71	15.5	10.6	29.7	22.9	42.6	36.5

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POWER

Condemn the Old Boiler

It is not often that the public exercises much concern as to the condition of steam boilers used in power plants, unless one of them explodes and kills somebody. Then censure is handed out right and left, and the less that is known about boiler practice the more harsh the criticism.

In the instance of a municipally operated electric-light plant those in charge decided that the lap-joint boilers were not fit for further service and new butt-joint boilers were accordingly installed in their place.

Before the old boilers were discarded, however, the taxpayers rose up in their might and loudly denounced the wastefulness of the town fathers for throwing out good boilers and wasting public money in purchasing new ones. In justification of their stand against this *needless* waste, a neighboring manufacturing plant was cited, where several boilers were in operation every working day of the year, and none of these boilers had been in service less than twenty years.

From a layman's standpoint these town folk were right in their views, and they were somewhat justified in taking the stand that if a set of boilers over twenty years old were good enough to serve a prosperous manufacturing company, their boilers, which were no older, were good enough to operate in the town lighting plant.

These town people were not aware of the fact that from the time a boiler is installed it begins to deteriorate, and that there comes a time when it is no longer safe to operate at the pressure originally allowed. Just when this time arrives is a matter of judgment on the part of those to whose care the boiler is intrusted.

Notwithstanding the opposition, the old boilers were thrown out and after the new boilers were installed and put to work the matter was dropped, except for muttered grumbling on the part of a few taxpayers.

Two years passed, and one day the air was rent with a mighty explosion; two men were killed and the boiler house of the manufacturing plant was completely wrecked. The old boilers that the town people had held up as safe and sound were lying in a heap of bricks and débris and one—the one that exploded—was a bent and twisted wreck.

Not one of the town folk can be found today who will intimate, let alone charge, that the town officials were wasteful of public money, or that they did not use the best of judgment when they threw out the old boilers.

There are but two reasons why an old boiler is allowed to remain in service: Either the firm owning it does not care to spend the necessary money for a new one, or the old boiler is made to do service while the new one is being installed, and it sometimes happens that the old boiler explodes in the meantime.

This matter of discarding old boilers will doubtless be allowed to rest with the judgment and conscience of the owner, in the future as in the past, a few States and cities excepted, until the death list becomes so great as to frighten owners into replacing their old boilers with new ones and, perhaps, convince the State legislature that every lap-seam boiler over ten years old is in reality a hidden mine, liable to explode sooner or later and add its toll of dead and wounded to the annual massacre.

Strength of Wheel Rims

Twelve or fourteen years ago, Prof. C. H. Benjamin presented papers on the "Bursting of Small Cast-iron Wheels" before the American Society of Mechanical Engineers which were remarkable in two ways: First, because he brought out complete and authoritative evidence on the subject, and, second, because of the small attention paid to them by engine builders. Some manufacturers today continue to place the flange between the spokes, just as they used to do, and they tell you with one breath that the strain in the rims of their wheels never exceeds one thousand pounds per square inch, and with the next breath that the efficiency of the flange joint is one hundred per cent. The professor stated in his paper that one would not think of putting a joint in the middle of a girder. This, we can understand, but why do not these engine builders understand it? It is not that they do not understand that it is poor design to put a joint in the middle of a girder which is carrying a transverse load, but rather that they do not believe the girder is loaded in that manner; that instead of the rim of a pulley between two arms being loaded like a girder, it is in tension, and, therefore, if the flange has enough metal in it, and the bolts are as strong as the flange, the efficiency of the joint would be one hundred per cent.

In this discussion, we will confine ourselves to pulley wheels, by which we mean wheels with thin rims used to transmit or receive power by belting or ropes. Regular flywheels, having rims of heavy rectangular cross-section, give little opportunity for bending between the arms and, besides, in such wheels the rim is sufficiently deep to permit the use of links for fastening the sections together.

The assumption that the rim of a pulley wheel is in tension only, due to the centrifugal force, is wrong, unless the rim is not attached to the spokes—is free to expand under the influence of the centrifugal force and assume a truly circular form. There are a few such wheels, in which the arms fit into a socket in the rim, and as the wheel revolves the rim can increase in diameter. Such a wheel is said to have a "free" rim, and such a rim is in tension only. It is clear that if in such a rim is put a flange having no weight, the section of which is designed to withstand the tension due to centrifugal force, such a flange can have an efficiency of one hundred per cent. But the flange must have weight; and so, even with a wheel with a "free" rim, the efficiency must be less than one hundred per cent.

If, now, we assume a pulley wheel in which the rim is attached to arms which are absolutely rigid and will not stretch, the rim, when rotating, will tend to bow out between the arms and will act like a girder loaded with a uniformly distributed load.

So, we have two theoretical cases: one in which the rim is free to expand and take a truly circular form, in which case the rim is in pure tension and is not subjected to bending. In the other case, the rim is attached to rigid arms that will not stretch, in which case the strain is due wholly to the bending moment.

In practice, however, the pulley wheel is between these two extremes. The rim expands some and pulls out the arms, and though the arms stretch some, yet they pull in the rim, so it is not correct to consider the rim in tension only, or as a girder carrying a uniformly distributed load only, and, to complicate the matter further, we have a strain induced by the flange itself.

To determine the strength of an actual wheel, recourse must be had to the results of experiments, and for these we are indebted to Professor Benjamin. These experiments show that a wheel with a well designed flanged joint, which is placed between the arms, will rupture at about one-half the speed of a similar wheel with a solid rim. As the strain varies as the square of the speed, this means that the flange joint is only one-quarter as strong as the solid rim, or that its efficiency is only about twenty-five per cent.

The wheels upon which the experiments were made were only twenty-four inches in diameter, and some may say that these results would not apply to a wheel, say, sixteen feet in diameter; but there is no reason for such an opinion, because the efficiency is a ratio and not an absolute quantity, while the flanges were carefully made to scale from a larger flange in a larger wheel in actual use. If there is any difference, it would be in favor of the small wheel, on account of the thinner sections, and, therefore, the better casting.

Do not confuse the measurement of the efficiency of such a flange joint with the measurement of the efficiency of the joint in, say, a boiler shell. If we say, in referring to the latter, that the efficiency of a certain joint is seventy per cent., the rivet strength being high, we mean that thirty per cent. of the metal has been cut away and that only seventy per cent. remains, and it follows that the greater the pitch of the rivets (making up their area by increasing the number of their rows) the greater will be the efficiency. There are joints in boilers that have an efficiency as high as ninety-eight per cent., but a similar procedure cannot be followed with flange joints in a pulley wheel, where, as the joint is strengthened by the addition of metal, the centrifugal force of that same metal increases in the same ratio the strain it is called upon to bear. So, when we refer to the efficiency of a rim joint as twenty-five per cent., we do not mean that it contains only one-fourth the amount of metal in the rim, or that if put in a testing machine and pulled it would break at one-quarter of the load on the solid rim, but, rather, that the strain in it is four times as much as the strain in the rim. Take a wheel sixteen feet in diameter, running at one hundred revolutions per minute, which is equivalent to a rim speed of five thousand twenty-six feet per minute. If the rim is "free" and has no joint, the tension in the rim due to centrifugal force is seven hundred pounds per square inch. If, however, the rim is fastened to the arms, and there is a flange joint between the arms, whose efficiency is only twenty-five per cent., then the strain in the rim is twenty-eight hundred pounds per square inch, and that the factor of safety is low in such wheels is shown time and again by the short interval of time which elapses between the derangement of the governing mechanism and the moment the wheel goes to pieces. Wheels of reputable make have been known to stand only a few seconds of racing.

Talk to an engine builder who persists in the use of the interarm joint about the efficiency of the flanges in the pulley wheel which he builds and he will at once begin to talk about the importance of good design, good workmanship and careful foundry work. We have no desire to appear to slight these very important matters, but the point that we wish to make is that of two similar wheels, one with a solid rim fastened to the arms, and the other with the rim joined by flanges placed between the arms, the latter wheel may be but one-quarter as strong as the former.

Generating Power for the Navy

A recently issued report of H. I. Cone, engineer-in-chief of the United States Navy, says that designs have been prepared for battleships with water-tube boilers, fitted for the use of oil fuel and coal, the oil fuel to be used in conjunction with coal or independently, and designs for destroyers for water-tube boilers with oil fuel only.

A high-speed marine steam turbine with reduction gear is being installed in the collier "Neptune," now building at the works of the Maryland Steel Company, Sparrows Point, Md. She is to be a twin-screw vessel, displacing 19,360 tons with a speed of fourteen knots. Steam at a pressure of two hundred pounds will be supplied by three double-ended Scotch boilers to a Westinghouse-Parsons turbine on each shaft, each turbine developing about four thousand shaft horsepower at one thousand five hundred revolutions at full power. Between each turbine and its propeller shaft is to be interposed a Melville-McAlpine gear, reducing the propeller speed to 136 revolutions per minute.

Tests have been completed during the year at the Norfolk navy yard of nineteen representative types of internal-combustion engines for launches. Of this number nine proved to be fit for naval service.

Considerable progress has been made on shore in the development of bituminous producer-gas power plants. Owing to a lack of funds the Bureau has been unable to do its part in the development of the internal-combustion engine for large naval vessels. As stated in the Bureau's last annual report, we cannot afford to delay this development and the recommendation is renewed for authority to expend as much as $250,000 for the purchase and installation of an internal-combustion engine plant and an able collier or other suitable hull in the event that it is thought wise to experiment along this line.

The generally accepted belief in the safety of water-tube boilers seems to have received a severe jolt by the recent explosion in Brooklyn. The authors of text-books upon boilers will have to get out revised editions.

The tendency is in the direction of getting more service out of a given amount of boiler-heating surface than has been thought practicable. Shall it be by putting in more grate surface or by burning more coal per square foot of grate?

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Engineer's First Duty

What is the first duty of an engineer on taking charge of a new plant?

E. F. D.

He should learn the condition and arrangement of the plant, what it will do and what is required of it.

Pressure Due to Heat

From a tank on the roof two pipes lead to the cellar. One is quarter-inch and the other two inches in diameter. Will a pressure gage read the same on the bottom end of each pipe?

P. D. H.

Yes, the pressure per unit of area for the same hight of water will be the same regardless of the diameter of the pipe.

Volume before and after Compression

How far will the piston move in compressing air to a pressure of 100 pounds in the cylinder with no rise in temperature?

B. A. C.

As the volume of air compressed in the cylinder will be inversely as its pressure, the volume of the compressed air will be

$$14.7 \div 114.7 = 0.128$$

of the original volume and the piston will, neglecting clearance, leakage and the heating of the air, move

$$1 - 0.128 = 0.872$$

of the stroke in compressing air to 100 pounds gage pressure.

Exhaust and Inside Lead

What is the difference between exhaust lead and inside lead and how much exhaust lead should a valve have?

E. I. L.

Inside lead is the opening which the exhaust port has when the valve is in the middle of its travel. Exhaust lead is the opening which the exhaust port has when the piston is at the end of the stroke, and should be sufficient to allow full port opening early in the return stroke so that there will be no excess in the back pressure. It is sometimes customary to make the exhaust lead twice the steam lead.

Load on Braces

How can I know that each brace in a boiler carries its full load?

L. O. B.

By being sure that none are loose when the boiler is empty but that all are equally or nearly equally tight, as may be proved by sounding with a hammer.

Air Compressor Capacities

How may I find the required lift for a 2½-inch gas-compressor valve? Give a rule to find how many cubic feet a gas compressor, 32x60 inches, with five pounds pressure above the atmosphere in the suction pipe and seventy-five pounds discharge, will deliver at one stroke?

A. C. C.

A lift of one-fourth the diameter of the valve will give a discharge area equal to the area of the valve but it has been found that 75 to 80 per cent. of this lift is sufficient. The volume of gas compressed may be determined by the formula

$$\frac{14.7 + 5}{14.7} \times \textit{Piston displacement} = \textit{Volume compressed}$$

and the volume after compression and subsequent cooling to the original temperature will be the above quantity multiplied by $\frac{14.7}{75 + 14.7}$. The actual amount will probably not exceed 85 to 90 per cent. of the theoretical depending on the volumetric efficiency of the compressor.

Steam Engine Dimensions

In making a small engine, how shall I proportion the different parts to make it look right? I mean such parts as the shaft, piston rod, valve stem, crank pin, connecting rod, etc.

S. E. D.

Make the main bearing one-half the diameter of the cylinder and two diameters long, the crank pin one-half the diameter of the shaft and one and one-quarter diameters long, the piston rod one-fifth the diameter of the cylinder and the connecting-rod length three times the length of the stroke and the diameter of the necks equal to the diameter of the piston rod.

Air Compressor Operation

We have lately installed a cross-compound two-stage air compressor. The governor is set to stop the compressor at an air pressure of 95 pounds and the engine invariably stops on the center on the high-pressure side, and before the air pressure falls sufficiently for the low-pressure side to start the engine again the steam in the receiver condenses and the pressure falls so low that the engine will not start until live steam is by-passed to the receiver.

We thought of placing a pressure-reducing valve in the bypass pipe leading to the receiver, and set it at 20 pounds, the pressure carried in the receiver, and keeping the valve in the bypass open so that the receiver pressure could not drop below the regular amount carried; would this interfere with the proper working of the engine in any way?

W. W. W.

The governor should be so set that the engine will not stop when the desired pressure is reached but will "creep" past the center. If a reducing valve is placed in the bypass it should be set to maintain a pressure in the receiver which will be just sufficient to move the engine off the center when the air pressure falls below the desired pressure.

Uptake Temperature

If the pressure carried in the boiler is 100 pounds, what should be the temperature of the escaping gases?

B. F. T.

From 450 to 500 degrees Fahrenheit.

Air Pressure for Lifting Water

In an air lift with 200 feet submergence, what pressure of air will be required to raise water 50 feet?

A. L. W.

The pressure required to start the water will be

$$\textit{Submersion} + \textit{lift} \times 0.434 + 5 = \textit{pressure.}$$

Once started the air pressure will depend on the quantity of water delivered but will never be less than is required to support a column of water of a hight equal to the lift.

Pressure in Radiators

In a steam-heating system, is the pressure in the radiators farthest from the boiler the same as in those which are nearest?

P. I. R.

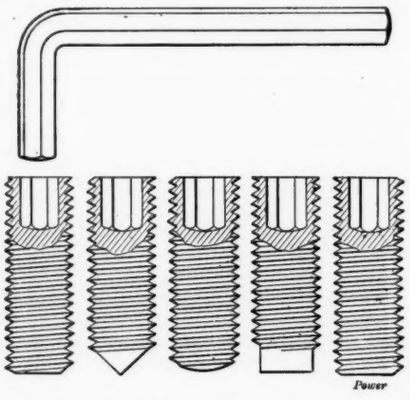
No. In order for steam or any other medium to flow from one region to another there must be a difference in pressure. In a one-pipe system the pressure is highest in the radiators in the upper part of the building.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news

The Allen Safety Set Screw

A safety set screw made from a solid bar of steel and guaranteed not to mushroom or upset in the hole has been recently placed on the market by the Allen Manufacturing Company, Hartford, Conn. These screws are made in a number of

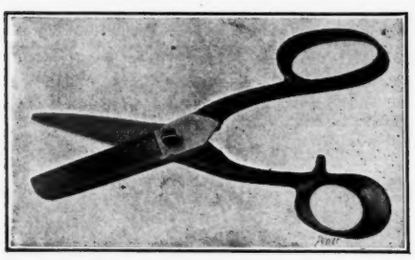


GROUP OF ALLEN SET SCREWS

different sizes ranging from ⅜ to 1 inch long and with a variety of points such as cup, conical, oval, dog and flat which are shown in the figure. A hexagonal hole formed in the other end serves as a hold for the wrench which can be made by bending a piece of hexagonal steel of the proper size at right angles, as illustrated in the above drawing.

Scissors for Belting, Packing etc.

The cut shows a pair of scissors for cutting leather, rubber, packing, linoleum, etc., which are being put on the market by Schuchardt & Schütte, 90 West street,



SCISSORS FOR BELTING AND PACKING

New York. The upper blade is a regular shear blade but with a longer handle than usual to give greater leverage.

The trouble with the ordinary scissors when cutting the materials mentioned is that the goods being cut is apt to be pushed along the blades instead of being cut. This is overcome in these scissors by having the edge of the lower blade serrated so that it prevents the material from slipping while the upper blade does the cutting.

These shears, known as "cogged scissors," are made in two sizes, 8½ and 11 inches respectively, and they will cut single-ply leather belting as easily as the ordinary scissors will cut cardboard.

The Stilwell Combination Water Heating and Softening System

In the illustration, Fig. 1, is shown a combination feed-water heater, filter and purifier built by the Platt Iron Works Company, Dayton, O.

This apparatus consists, as shown in Fig. 2, of a cast-iron heating chamber containing a system of pans over which the water and chemicals must pass, thereby thoroughly mixing the two and bringing them in direct contact with the exhaust steam. This heating chamber, which is fitted with an efficient oil separator, may be used either on the thoroughfare or induction principle. It is located on top of a large purifying and filtering chamber built of heavily ribbed cast-iron sections, with ground joints and permanent gaskets, and is designed to withstand ten pounds back pressure.

The system consists in using the heat

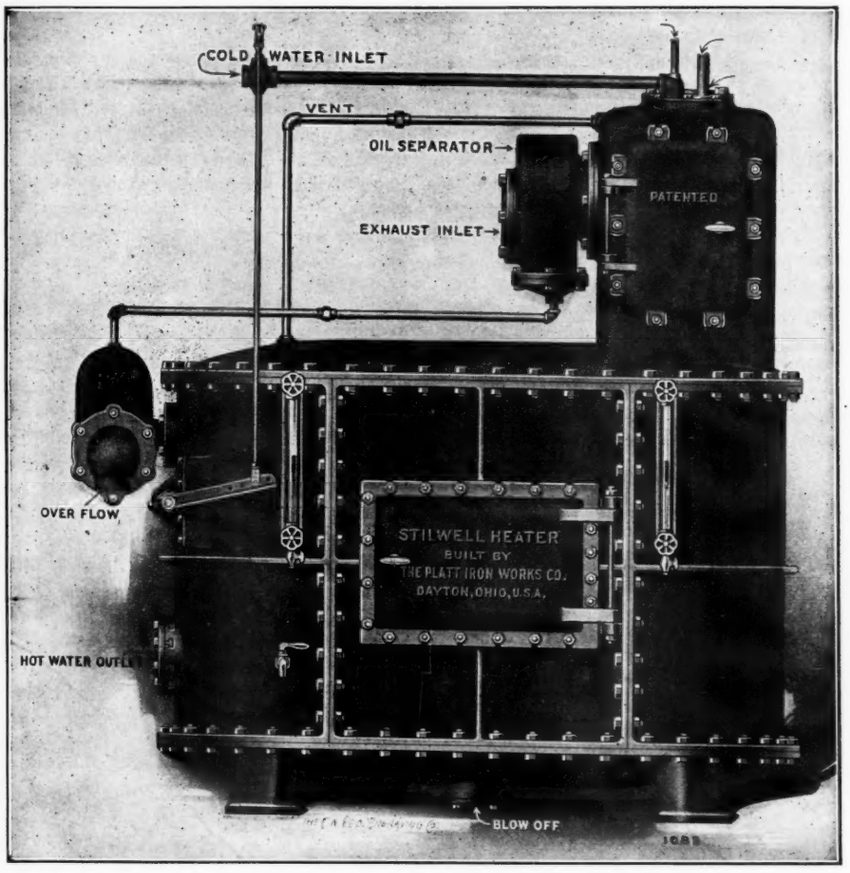


FIG. 1. EXTERIOR VIEW OF STILWELL HEATER

of the steam for removing the temporary hardness, such as carbonates, chemicals being used to remove permanent hardness caused by sulphates, chlorides, etc. In operation it is designed that the reagent will be fed continuously in proportion to the cold water, both entering the heating chamber at the same point, the reagents being handled by an auxiliary plunger on the feed pump.

After the water has passed through the heating chamber it reaches a settling chamber below, in which the greater portion of the impurities will settle to the bottom, from which they can be blown off. The water then passes upward through a blanket filter into the purified-water chamber from which the pump suction takes its supply.

A device recently designed and made part of this system consists of a water-forcing chemicals in proper proportion to the feed water, make up the system, which is automatic when once installed.

The "Change Blade" Screwdriver

A new screwdriver is being put on the market by Kinckiner & Scott, 626 North Twelfth street, Philadelphia, Penn.

The handle is of red brass in skeleton form, with a covering of mahogany. The blades are of tool steel with a tempered cross pin, which engages with a slot in the brass handle and takes the strain of the work. The end of the blade, which enters the handle, is threaded to fit the cap nut which retains it in position in the handle. The blades are made in 3-, 6- and 8-inch lengths and are easily interchanged.

A change-blade screwdriver is made

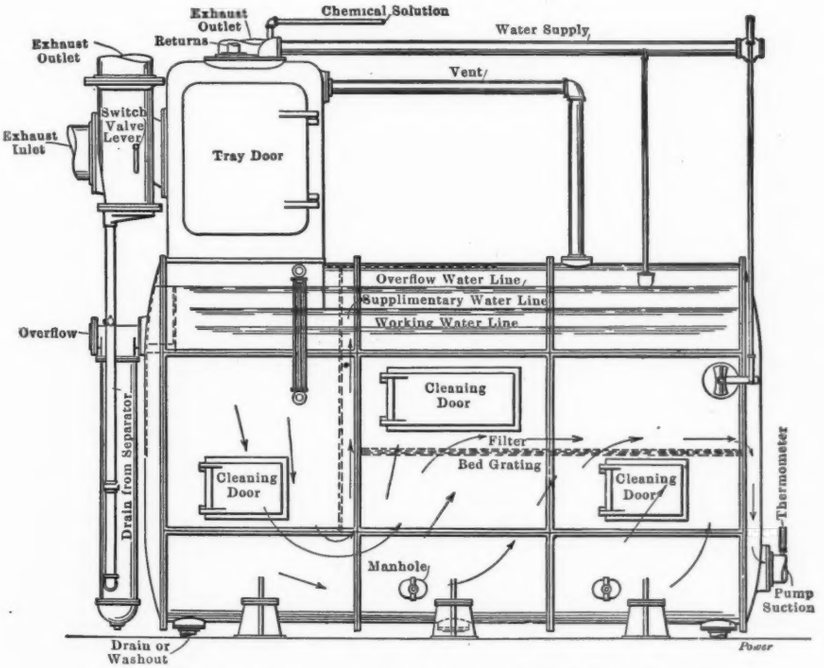


FIG. 2. VIEW OF HEATER, INDICATING INTERIOR CONSTRUCTION

sealed supplementary bypass which insures the delivery of hot treated water to the boiler-feed pumps in case the filter blanket, if neglected, should become clogged up. The bypass automatically operates when the water rises to the proper level in the heating and purifying chambers. This special feature, being water sealed, prevents any scum or floating particles from passing over into the purified-water chamber.

A skimmer is provided in the heating chamber for skimming the surface of the water, this skimmer also acting as a point of overflow into the trap. Large hinged doors are provided, permitting easy access to all parts and any internal part can be readily removed through these doors. Float valves, mixing tanks and auxiliary pumps for especially for electrical work. In it the entire handle and blade are insulated with the exception of the extreme end which engages the screw slot.

Congress of Refrigeration to Be Held in America

The executive committee of the American Association of Refrigeration held an important meeting at the Great Northern hotel, Chicago, on December 15, to perfect an organization and provide ways and means for the third international congress which the recent second congress in Vienna voted shall be held in the United States. Several members of the committee came from widely separated sections of the country. Rivalry for the privilege of entertaining the scientists who will represent the refrigerating interests of more than forty foreign countries is already active. New York, Washington, Philadelphia, Birmingham, St. Louis and Chicago are among the places advocated. Chairman Homer McDaniel, of Cleveland, was authorized to appoint a committee of five to prepare a budget of the necessary expenditures for conducting the congress, to decide upon the most suitable date and place for holding it, to prepare a general scheme of entertainment for the foreign delegates, and to report the results of their labors to the next regular annual meeting of the association. It was the general opinion of those present that the committee will without difficulty secure a sufficient fund to justify our Government in extending its invitation to foreign countries to participate in the congress by the appointment of official delegates, as was done by our Government for the Paris congress of 1908 and the Vienna congress of this year.

Brooklyn Engineers Hold Annual Dinner

The Brooklyn Engineers' Club held its annual dinner on December 15, 1910, at the Hamilton Club, Brooklyn; the fact that there was such a large call for tickets led the committee to arrange for greater accommodations than the club's own banquet room affords.

George A. Orrok, the retiring president of the club, who acted as toast master, briefly reviewed the work accomplished during the year.

The first speaker to be introduced was Dr. Rossiter W. Raymond, secretary of the American Institute of Mining Engineers. Doctor Raymond's talk was on the "Panama Canal" and was based on the information he obtained during the excursion which he and other members of the Institute of Mining Engineers recently made to the canal.

Dr. Alexander C. Humphreys, president of Stevens Institute of Technology, was next introduced. His topic was "Engineering Education: the Concrete Rather than the Abstract." He maintained that much valuable engineering knowledge can be learned only in the school of experience.

Major John F. O'Rourke, president of the O'Rourke Engineering and Contracting Company, spoke on "After-dinner Engineering."

After the dinner, the following officers of the club for 1911 were elected: President, Major Winfred H. Roberts; vice-president, John M. Steinmetz; secretary, Joseph Strachan, reëlected; treasurer, William T. Donnelly; directors, H. C. Keith and W. F. Wells.

There were about 160 members and guests present.

Steam Pipe Bursts in Lowell

It is reported in the daily press that on December 24 a steam pipe burst in the Perry street power plant of the Lowell (Mass.) Electric Light Company. Six men were injured and fragments of the pipe damaged the brickwork of the building to some extent. The plant was shut down for an hour.

OBITUARY

Rudolph Wolf, founder of the Great Engineering Works at Magdeburg-Bugkau, inventor of the Wolf compound engine and identified with the early use of superheated steam, died on the twentieth of November in his seventy-ninth year.

Matthew Kennedy, treasurer of the Kennedy Valve Manufacturing Company, Elmira, N. Y., died on November 26 at his home in Coxsackie. He was born in Ireland in 1840, and with his brother Daniel established the business with which he was so long identified.

BOOKS RECEIVED

DYNAMO ELECTRIC MACHINERY. By Samuel Sheldon. D. Van Nostrand Company, New York. Cloth; 328 pages, 5x7½ inches; 210 illustrations; indexed. Price, $2.50.

ELECTRICITY EXPERIMENTALLY AND PRACTICALLY APPLIED. By S. W. Ashe. D. Van Nostrand Company, New York. Cloth; 344 pages, 5x7½ inches; 422 illustrations; indexed. Price, $2.

BROOKES AUTOMOBILE HANDBOOK. By L. Elliott Brookes. Frederick J. Drake & Co., Chicago, Ill. Leather limp; 701 pages, 4x6½ inches; 320 illustrations; tables; indexed. Price, $2.

THE CONSTRUCTION AND WORKING OF INTERNAL COMBUSTION ENGINES. By R. E. Mathot. D. Van Nostrand Company, New York. Cloth; 554 pages; 5½x9¼ inches; fully illustrated; indexed. Price, $6.

DESIGN OF MARINE MULTITUBULAR BOILERS. By James D. McKnight and Alfred W. Brown. The Technical Publishing Company, Ltd., and D. Van Nostrand Company, New York. Cloth; 48 pages, 6x10 inches; illustrated; indexed. Price, $1.50.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

WATER WHEEL. Arnold Pfau, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 978,335.

ROTARY ENGINE. John W. Larimore, Benton, Ill. 978,602.

WAVE MOTOR. Thomas Nixon, Santa Barbara, Cal. 978,628.

ROTARY ENGINE. Samuel Haudenshield, Carnegie, Penn. 978,743.

BOILERS, FURNACES AND GAS PRODUCERS

STEAM GENERATOR. James J. Bush, New York, N. N. 978,135.

SMOKE CONSUMER. William McArdle, Montreal, Quebec, Canada, assignor to the Perfect Simplex Combustion Company, Montreal, Canada. 978,467.

SHAKING AND DUPMING GRATE. Chas. F. Hutchinson, Kingsville, Md., assignor to Hutchinson Bros., Kingsville, Md., a Corporation. 978,589.

GENERATOR AND SUPERHEATER. John G. Massie, East St. Louis, Ill., assignor to the Massie Generator and Radiator Company, East St. Louis, Ill., a Corporation of Illinois. 978,769.

OIL BURNER. John R. Pring, Shawnee, Okla. 978,780.

CRUDE-OIL BURNER. Emory A. Wales, Oklahoma, Okla. 978,797.

POWER PLANT AUXILIARIES AND APPLIANCES

GAGE COCK. Charles Wright, Youngwood, Penn., assignor to the Wright Specialty Manufacturing Company. 978,256.

ENGINE-STARTING DEVICE. Peter P. Au Buchnon, French Village, Mo. 978,264.

VALVE. John William Harkom, Melbourne, Quebec, Canada. 978,288.

AUTOMATIC CUTOFF VALVE. Francis Hodgkinson, Edgewood Park, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 978,294.

BOILER-TUBE CLEANER. William L. Miggett, Ann Arbor, Mich., assignor to Raphael Herman, Detroit, Mich. 978,326.

ROTARY PUMP. James Baguley, Evanston, Wyo. 978,350.

CONDENSER. Royal D. Tomlinson, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 978,411.

VALVE AND VALVE-OPERATING MECHANISM. Fred Loedige, Chicago, Ill. 978,463.

OIL CUP. Verner J. Wahlstrom, New York, N. Y. 978,521.

PISTON-ROD STUFFING BOX AND LUBRICATOR. Walter McLain, Spiritwood, N. D. 978,611.

HOSE COUPLING. Bernard Morgan, Newport, R. I. 978,619.

PUMP. Garl Nicholls, McFall, Mo. 978,626.

PUMP. Edwin E. Slick, Pittsburg, Penn. 978,668.

CONDENSER. Evi W. Christie, Sewaren, and Tom Roberts, Roselle Park, N. J., assignors to Wheeler Condenser and Engineering Company, Carteret, N. J., a Corporation of New Jersey. 978,697.

STEAM TRAP. Vernon Bradley Convis, Toronto, Ontario, Canada. 978,701.

VALVE. James E. Davidson, Butte, Mont. 978,706.

ROTARY PUMP. Michael E. Durman, Detroit, Mich. 978,715.

GRAVITY VALVE CAGE AND VALVE FOR PUMPS. Jesse B. Garber, Salem, Ohio, assignor to the Deming Company, Salem, Ohio, a Corporation of Ohio. 978,729.

VALVE GEAR. Hiram P. Graves, Elmira Heights, N. Y. 978,737.

VALVE. Joseph Huebsch, Milwaukee, Wis. 978,752.

CENTRIFUGAL PUMP. Joseph Hurst, Louisville, Ky. 978,753.

CHECK VALVE. Jonathan Johnson, Lowell, Mass. 978,757.

DEFLECTOR FOR SMOKE-BOX SUPERHEATERS OR FEED-WATER HEATERS. Samuel M. Vauclain, Philadelphia, Penn., assignor, by mesne assignments, to Baldwin Locomotive Works, Philadelphia, Penn., a Corporation of Pennsylvania. 978,795.

LUBRICATOR. Carl Roberts Briggs, Ravenna, Ohio. 978,819.

ELECTRICAL INVENTIONS AND APPLICATIONS

APPARATUS FOR ELECTRIC SMELTING. Frank Creelman, New York, N. Y., assignor to the Wilson Carbide Works Company of St. Catharines, Ltd., St. Catharines, Canada, a Corporation. 978,137.

ELECTRIC MOTOR-CONTROLLING APPARATUS. Harry Ward Leonard, Bronxville, N. Y. 978,173.

COMBINED SWITCH SOCKET AND PLUG. William Pinkney McNeel, San Antonio, Tex. 978,322.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., Col. E. D. Meier; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power

A week or so ago we attended a meeting of the Technical Publicity Association—an association of advertising managers of machinery and allied concerns.

The subject of the evening was "copy," with especial reference to advertising copy in technical papers.

What was done to technical paper advertising copy was good and plenty—and then some.

The speakers took some horrible examples, hung 'em on a gibbet, stood off about two feet with a big blunderbuss, closed their eyes and banged away.

By the perverse fate which seems to pursue technical advertising, not one of the three speakers was a man conversant with the subject.

One of them admitted the fact—talked about something else—and made a good speech.

But to begin with each man felt that he must call attention to some atrocious advertisement from a technical paper and use *that* as an example of the average.

Wrong again, gentlemen; set your sights a trifle over; there's a strong wind blowing.

We can take any general monthly magazine published and show you just as poor ads as you can find in a technical paper.

And as good ones in the latter as you'll find in the former.

Such a statement will make the captious critic stiffen up and curl his Gilletted lip over his Colgated teeth—but the proof is easy.

The trouble is that the average advertising man will skim over the advertising pages of a technical paper and, failing to find the *art* work and the *variety* of subjects that help to make the pages of a general magazine attractive, will pronounce the technical advertising poor.

He misses the Corset lady and the Smooth Cream blonde; he regrets the absence of the Lightning Auto and the visions it conjures of joy-rides on Easy Street; he looks in vain for the work of the copy-writer, who, mounting his untamed vocabulary, jabs it a couple of times with his favorite adjective, and is off down the line in a mess of words.

The purpose of technical advertising is to give technical information that will interest the reader of the paper who is also a possible buyer of the article.

This cannot be accomplished by word trickery and frilly illustrations.

It must be done simply and sincerely, and earnestly and clearly.

You may take any issue of this paper and you'll find poor ads. and good ones.

The poor are distinguished by the size of their claims with a corresponding lack of information. They are either mere business cards or they consist mostly of claiming to be the "most durable, efficient and economical on earth."

But the *real* advertisements, the kind you'll read and be interested in, are those that tell you something and prove it.

And these it will pay you to read and act upon.

It must be remembered that the small ad. is necessarily limited in what it can show and say.

Some advertisers must get more information into your hands before they can prove their case, and this they gladly furnish on request.

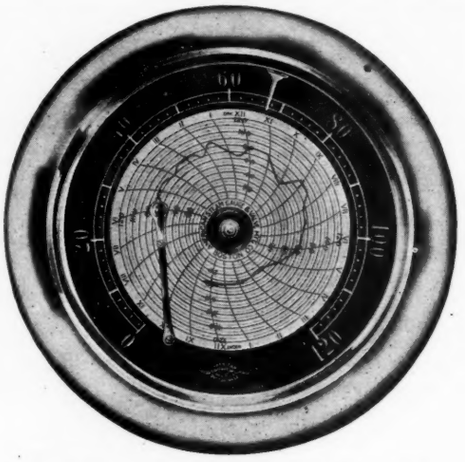
It's like the story of the freshman in his first trigonometry class: The instructor asked, "Mr. Smith, have you proved that proposition?"

"Well, sir," was the reply, "'Proved' is pretty strong word, but I may say I have rendered it highly probable."

When an advertiser makes his proposition "highly probable" in his advertisement and you are interested, ask him to send you the whole story.

Ten to one it will be well worth getting.

American Combined Pressure And Recording Gauge



The American Pressure and Recording Gauge.

Tells You What's Going On

You may not be able to watch the fireman all day—you've got other and more important work to do.

You can't stand over him and see that he feeds the right quantity of coal all the time, that he keeps up the pressure.

But by means of this valuable pressure and recording gauge you can tell any time just exactly what has been done.

You can tell whether the boiler has been neglected and when—

You can see at a glance what pressure there was on the boiler at 10 o'clock and what pressure at 12.

And furthermore you will have evidence of "able engineering" in such shape that you can show and explain to the boss.

In fact the Pressure and Recording Gauge can be installed right in the manager's office if desired so *he* can see that everything is running smoothly in the power plant.

It's a power plant necessity, a device that prevents shirking, that assures the *right* kind of power plant service.

A line to us will bring complete details of this and our other steam specialties

American Steam Gauge & Valve Mfg. Co.

Boston, Mass.

New York, 26 Cortlandt St. Atlanta, 525 Candler Bldg. Chicago, 130-132 Jefferson St. Pittsburg, Columbia Bank Bldg. San Francisco, Monadnock Bldg. Montreal, 444 St. James St. Los Angeles, 213 So. Los Angeles St.

How Many Indicators Will You Ever Buy?

It depends solely upon the good judgment you use when you first buy an indicator.

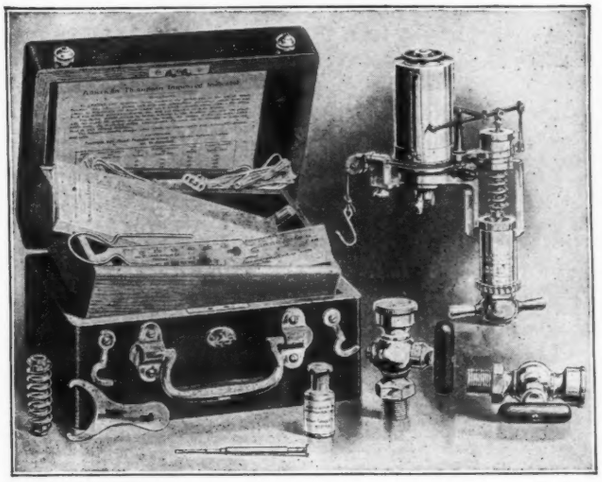
If you are led into the purchase of an indicator because it is cheap, you'll find that cheapness in indicators is the most expensive experimenting you ever tried.

If you say to yourself—"This instrument must last me a life time, it must be absolutely accurate today, tomorrow and always, else it is worthless"—then you'll forget the few dollars' difference between the best and near-best and buy an American-Thompson Improved Indicator.

Then you'll need to buy only one Indicator. The American-Thompson Indicator costs $55.00 which can be paid in easy terms—$5.00 first payment and then $5.00 monthly for the balance and it is worth every cent of its price.

It is sold with no premiums, no fancy incidentals, no anything but absolute "indicator merit," which assures accurate diagrams to you. It gives you real help, that kind of help that enables you to improve your conditions, to advance your salary, to make yourself a better engineer.

And that kind of an instrument, no matter what its cost, is the least expensive instrument you can buy.



Buy An American-Thompson

Because it is the Best Indicator Made. Special materials are used to begin with. Experienced men make the parts and experienced men put them together.

The detent motion is an improved motion; it is patented and can be used only on the American-Thompson.

It is positive, gentle and instantaneous.

It engages and releases the drum on the highest speeds as rapidly as the hand can move and is absolutely free from jars and shocks.

And each instrument before shipment is put to a final test, more severe than any you will ever give it.

Consequently when you buy an American-Thompson you buy a lasting instrument, one that you can use year after year knowing that the diagrams you take with it are right. Mail us the first payment ($5.00) today and we'll ship you the Outfit at once, with the privilege of examining it for a period of five days.

You risk nothing—so act on this matter today—you'll be the gainer.

If you want additional information send the coupon for our Indicator Proposition No. 91. But don't put this off until tomorrow, do it now.

American Steam Gauge & Valve Mfg. Company

Boston, Mass.

New York Chicago Atlanta
Pittsburg San Francisco Los Angeles

The Outfit

The Indicator, inclosed or exposed spring type as preferred, two single cocks or one 3-way cock (as preferred), one scale, all necessary wrenches, screw-driver, bottle of porpoise oil, etc., all inclosed in a neat mahogany box, together with 200-page book—Indicator Instructions.

The Terms

Suitable references and an initial payment of $5.00 secure the Indicator Outfit at once. $5.00 a month pays the balance. The total cost is $55. Remember that, as stated, you may have the outfit for examination without risk whatever.

American Steam Gauge & Valve Mfg. Co.
Boston, Mass.

Send me full details of your Indicator Offer No. 91.

Name . . . . . .

Address . . . . . .

City and State . . . . . .

BUSINESS ITEMS

The Standard American Dredging Company, of San Francisco, has just ordered a 700-horsepower alternating-current induction motor from the Westinghouse Electric and Manufacturing Company.

The Simplex Engineering Company, 1111 Locust street, Philadelphia, Penn., has received a letter from Charles R. Weaver, chief engineer of the Markeen hotel, Buffalo, N. Y., in which he says: "The 6-inch Simplex pipe joint clamp which I ordered from you through Root, Neal & Co., arrived O. K. and I wish to state that I put the clamp on a leaky joint that had been giving us very much trouble and am happy to say that our troubles are over as far as that joint is concerned. Other clamps had failed to hold, but the Simplex is doing all you claimed for it, and has saved us a bill of over $80 besides the inconveniences of a shutdown and possibly the price of a new 6-inch valve in addition to that. The Simplex puts the packing ring just where it belongs (tight against the leaky joint) only, not squeezed out sideways as with other clamps."

The Murphy Iron Works, Detroit, Mich., have recently secured an order from the American Sugar Refining Company for Murphy furnace equipment for twelve 500-horsepower Babcock & Wilcox boilers to be installed in its new power house, at Boston, Mass. This is to be one of the most complete plants of its kind in the United States, and the decision regarding mechanical stokers was the result of the results being obtained by the Murphy furnaces installed in connection with 36 Babcock & Wilcox boilers at its refinery at Chalmette, La., which is one of the largest refineries in the United States. The furnaces were installed in this plant after a thorough investigation extending over a period of 14 months, and the results obtained are such it felt justified in placing its order with this company for the equipment of its Boston plant. The Murphy Iron Works also advise that the year 1910 has shown the largest volume of business received by this company since its organization, 33 years ago, and the prospects are excellent for the coming season. Within the last 60 days, 26 orders for Murphy furnace equipment have been received.

One of the largest contracts for pumping machinery ever placed has just been awarded to the Alberger Pump Company, of New York City, by the Bureau of Yards and Docks, the total sum involved being slightly over $323,000. The contract covers pumping equipment for the three new dry docks to be constructed by the Government at New York, Puget sound and Pearl Harbor navy yards, and includes all told eleven 54-inch vertical volute pumps, each direct connected to a 550-horsepower induction motor ,and seven 15-inch vertical volute drainage pumps, each direct connected to an 85-horsepower induction motor; also all necessary suction and discharge piping, electrically operated gate valves for the same and all electrical controlling apparatus for the motors. Three of the 54-inch units will be located at New York, while four of these units are required for each of the other docks. Each unit is required to operate against a static head, varying from zero when the dock is full to a maximum of 42 to 44 feet when the dock is completely empty, while operating at a constant speed of 219 revolutions per minute without exceeding the rated horsepower of the motor at any point. In addition to this, each pump is required to maintain an average capacity of 66,000 gallons per minute when emptying the dock from approximately mean high water to one foot above the elevation of the keel blocks, the static head varying from zero to 32 or 34 feet. Under these conditions the contractor has guaranteed an average efficiency of 45 per cent. for New York and Pearl harbor and 46 per cent. for Puget sound, these efficiencies being the ratio of the actual useful work in pumping the docks to the electrical input to the motors, and consequently includes all losses in the motors, pumps and piping. To meet these average efficiencies it is necessary for the pumps, themselves, to reach a maximum efficiency of nearly 80 per cent. In addition to obtaining a high efficiency, the other characteristics of the pump have to be carefully determined in order to meet the special conditions involved in this class of work.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

WANTED—Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," POWER.

ENGINEER-FIREMAN wanted; wages $17.50 per week; competent to handle Corliss engine; send references. "Manufacturer," POWER.

AN ENGINEER in each town to sell the best rocking grate for steam boilers. Write Martin Grate Co., 281 Dearborn street, Chicago.

WANTED—Technical writer to prepare catalogs, circulars and advertisements for power plant apparatus and machine tools; state salary; opportunity for capable young man. Address Box 344, POWER.

DESIGNER on direct current motors and generators; must be capable of taking charge, and able to design special requirements in the way of windings on standard frames, interpole motors, etc., without guess work or experimenting; only experienced men need apply, giving full particulars and salary required to Box 345, POWER.

SITUATIONS WANTED

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

OILER wants position; thoroughly familiar with power plant work. Box 343, POWER.

MECHANICAL ENGINEER, technical man, experienced in the development and design of modern manufacturing and power plants. Box 342, POWER.

MECHANICAL ENGINEER, with three years' experience in gas and steam engine operation and boiler house economy, wishes a position with manufacturers or operators. Box 339, POWER.

ENGINEER WANTS POSITION in a plant where safety, highest efficiency and soberness are appreciated; have had five years' experience as chief engineer; thoroughly competent; best references. Henry DeBerry, 410 Broadway Ave., Little Rock, Ark.

A highly efficient mechanic with technical education and inventive ability, wishes position taking charge of experimental shop, either with work of purely mechanical or electrical character. Box 337, POWER.

POSITION WANTED by engineer of twenty-eight years' experience, stationary and marine practice; hold New York second class license, first class American and British certificate; want position in charge of manufacturing or hotel plant; will go anywhere; first class references. Box 340, POWER.

WANTED—Position by first class engineer; have had 17 years' experience in power house work; understand dynamos and motors; have been chief in light and power plants for 10 years; am married; 38 years old and of good habits and willing to work for interest of employer; state wages in first letter; understand producer gas engines also. Box 341, POWER.

MISCELLANEOUS

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

PATENTS secured. C. L. Parker, Solicitor of Patents, 4 McGill Bldg., Washington, D. C.

IF YOU DESIRE to lern the latest improvements in steam boilers, correspond with the Detroit Water Tube Boiler Co., Detroit.

WANTED—Agents, engineers preferred, to sell boiler cleaning chemicals; liberal commissions paid. Continental Chemical Co., Forest Bldg., Philadelphia, Penn.

PATENTS—H. W. T. Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make investigations and report if patent can be had and exact cost.

EVERY ENGINEER should be posted regarding the new system of vacuum heating installed without payment of royalty; I have valuable information; write today. T. L. Reeder, 1417 W. Jackson Blvd., Chicago, Ill.

ENGINEER AND FIREMEN—Send 10 cents in stamps for a 40-page pamphlet containing a list of questions asked by an examining board of engineers. Stromberg Publishing Co., 2703 Cass Avenue, St. Louis, Mo.

WE WANT the agency of American manufacturers of goods that are required by machinists and engineers; we are importers of American engineering specialties and manufacturers' sole agents for Austria, Hungary and Balkan states, and have ample show rooms and offices. John Kollin & Co., General Office, Budapest, Hungary.

THE ANNUAL MEETING of the stockholders of the Hill Publishing Company, for the election of directors for the ensuing year and for the transaction of such other business as may properly come before the meeting, will be held at the offices of the company, in the Hallenbeck building, 497-505 Pearl street, Borough of Manhattan, New York City, N. Y., on Tuesday, January 31, 1911, at 12 o'clock noon.

Dated, New York City, December 9, 1910.

Robert McKean, Secretary.

FOR SALE

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

FOR SALE—Small screw cutting lathe, chuck and tools; fine condition; bargain. South Bend Machine Co., South Bend, Ind., 424 Madison St.

FOR SALE—20x48 Wheelock engine and two 72"x18' high pressure tubular boilers in good condition cheap. Address "Engineer," Box 2, Station A, Cincinnati, Ohio.

CONDENSER FOR SALE—Blake vertical twin air pump and jet condenser; size 14x38 x21; pistons metallic packed, brass fittings, perfect condition, used four years; we need larger condenser. The Stanley Works, New Britain, Conn.

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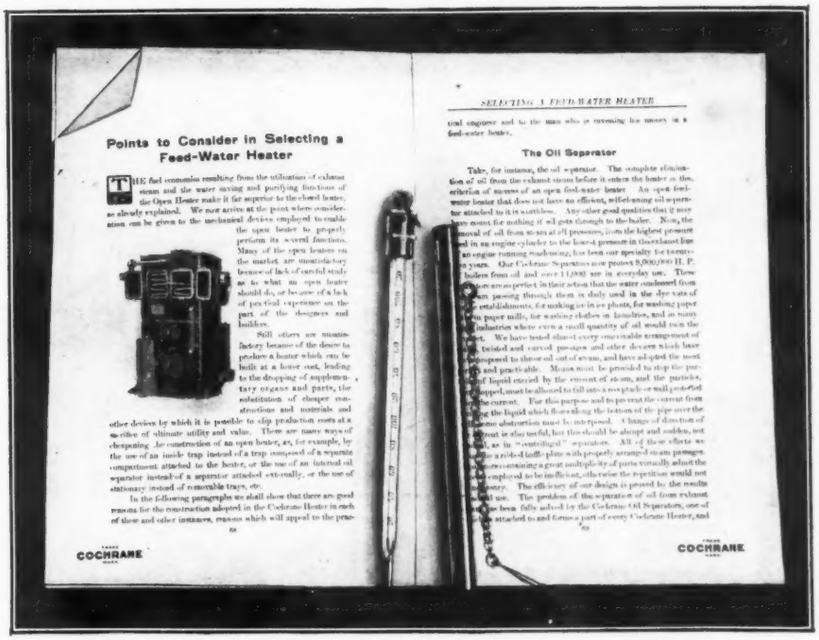

Points to Consider in Selecting a Feed-Water Heater

THE fuel economies resulting from the utilization of exhaust steam and the water saving and purifying functions of the Open Heater make it far superior to the closed heater, as already explained. We now arrive at the point where consideration can be given to the mechanical devices employed to enable the open heater to properly perform its several functions. Many of the open heaters on the market are unsatisfactory because of lack of careful study as to what an open heater should do, or because of a lack of practical experience on the part of the designers and builders.

Still others are unsatisfactory because of the desire to produce a heater which can be built at a lower cost, leading to the dropping of supplementary organs and parts, the substitution of cheaper constructions and materials and other devices by which it is possible to clip production costs at a sacrifice of ultimate utility and value. There are many ways of cheapening the construction of an open heater, as, for example, by the use of an inside trap instead of a trap composed of a separate compartment attached to the heater, or the use of an internal oil separator instead of a separator attached externally, or the use of stationary instead of removable trays, etc.

In the following paragraphs we shall show that there are good reasons for the construction adopted in the Cochrane Heater in each of these and other instances, reasons which will appeal to the practical engineer and to the man who is investing his money in a feed-water heater.

COCHRANE

SELECTING A FEED-WATER HEATER

The Oil Separator

Take, for instance, the oil separator. The complete elimination of oil from the exhaust steam before it enters the heater is the...erfection of success of an open feed-water heater. An open feed-water heater that does not have an efficient, self-cleaning oil separator attached to it is worthless. Any other good qualities that it may have count for nothing if oil goes through to the boiler. Now, the removal of oil from steam at all pressures, from the highest pressure used in an engine cylinder to the lowest pressure in the exhaust line of an engine running condensing, has been our specialty for twenty-two years. Our Cochrane Separators now protect 8,000,000 H. P. of boilers from oil and over 11,000 are in everyday use. These Separators are perfect in their action that the water condensed from steam passing through them is daily used in the dye vats of textile establishments, for making ice in ice plants, for washing paper stock in paper mills, for washing clothes in laundries, and in many other industries where even a small quantity of oil would ruin the product. We have tested almost every conceivable arrangement of baffle, twisted and curved passages and other devices which have been proposed to throw out oil of steam, and have adopted the most efficient and practicable. Means must be provided to stop the particles of liquid carried by the current of steam, and the particles, when stopped, must be allowed to fall into a receptacle or well protected from the current. For this purpose and to prevent the current from picking up the liquid which flows along the bottom of the pipe over the...some obstruction must be interposed. Change of direction of current is also useful, but this should be abrupt and sudden, not gradual, as in a "centrifugal" separator. All of these effects we obtain in a right baffle-plate with properly arranged steam passages. Separators containing a great multiplicity of parts virtually admit the oil employed to be inefficient, otherwise the repetitious would not be necessary. The efficiency of our design is proved by the results of actual use. The problem of the separation of oil from exhaust steam has been fully solved by the Cochrane Oil Separators, one of which is attached to and forms a part of every Cochrane Heater, and

COCHRANE

HOW HOT?

"How hot?" is the principal question that the steam engineer has to answer.

How hot is the water entering the boiler? It should never be much below 210° F., nor need it be if it is heated in a COCHRANE OPEN FEED WATER HEATER receiving the exhaust of auxiliaries, as pumps, fan and stoker engines, etc. If the water has been heated to 210° F. by spraying through the steam-bath of the Cochrane Heater, all air and gases, which are essential to corrosion, will have been driven off, and the driving off of carbon dioxide gas will have resulted in the breaking up of bi-carbonates into the insoluble mono-carbonates, which rapidly settle out of the water, thus keeping that much scale out of the boiler. While the water is hot, it can be easily and conveniently treated with soda ash or other reagent to correct permanent hardness, as in the SORGE-COCHRANE HOT PROCESS WATER SOFTENING SYSTEM.

How hot is the exhaust steam passing to the condenser? This is a better gauge of the vacuum than most vacuum gauges, provided the thermometer be inserted at the steam inlet.

How hot is the tail-water or hot-well water from the condenser? If it is a modern condenser, this should not be more than 10 or 15° below the temperature corresponding to the vacuum. Any greater difference indicates imperfect action of the condenser.

How hot is the steam in the steam main? This, by comparison with the pressure gauge, will show how much superheat there is.

By determining **how hot the steam is after it has been wire-drawn through a throttled opening,** the amount of moisture is readily determined, that is, you can make your own calorimeter and appreciate the need of a COCHRANE STEAM SEPARATOR to keep the moisture in the steam out of the engine or turbine

How hot is your lubricating oil when it will catch fire from a match, held over it? This is the flash point, an important thing to know when buying oil.

How hot are the armature and fields of the generator, the switches on the switchboards, and a thousand other things about a power plant?

"How hot" can be answered if you have one of these first-class improved Cochrane thermometers. This thermometer is fitted with a nickel-plated, metal carrying case, with a chain and safety-pin, and has been especially designed for the use of the stationary engineer. **For a limited time it will be sent, together with a copy of our treatise on the**

"Profitable Utilization of Exhaust Steam,"

for SEVENTY-FIVE CENTS ($ .75). You could not buy one in the open market at anything like this price. **This offer is open to only one operating engineer in each plant. The name of the plant and the kind of feed water heater now in use must be stated.** The book is thrown in so that you may be able to make profitable use of the information about temperatures that you may obtain with the thermometer.

Harrison Safety Boiler Works

17th and Clearfield Sts. Philadelphia, Pa.

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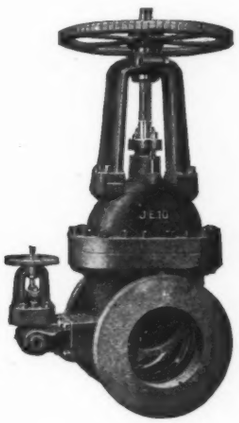
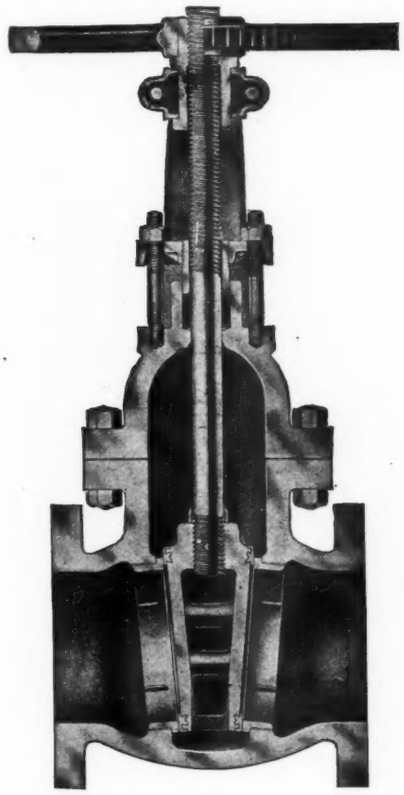
For The High Pressures Use Jenkins Bros. Extra Heavy Iron Body Gate Valves





THE highest steam pressures used in modern power plant service will not affect these Jenkins Bros. High Pressure Gates. They are made substantially and hold their own under the severest conditions.

All parts are renewable and interchangeable. The stuffing boxes can be repacked while the valves are wide open and under steam pressure. All flat surfaces are eliminated assuring great strength.

They are of the double face, solid-wedge type, have removable seat rings, and bronze faced wedge.

They are made with either inside screw, stationary spindle, or outside screw and yoke, rising spindle. They are also furnished with or without by-pass.

Write for catalog fully describing all our Jenkins Bros. Valves, Jenkins '96 Packing and other specialties.

Jenkins Bros.

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